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FINAL REPORT, KEL-F LINER FOR THE ATLAS MARK 4 SUSTAINER TURBOPUMP

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FOREWORD

This report was prepared under G.O. 8333 and Contract AF04(694)-135, Part I, Item 1, as amended by Request for Service Order 135-62-3 and Amendments No. 1 and 2 thereto.

ABSTRACT

Presented are the results of a program to develop, evaluate, and qualify a Kel-F liner for the oxidizer inlet passage of the Mark 4 sustainer turbopump, and to measure turbopump shaft deflection during operation.

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INTRODUCTION

Four Mark 4 LOX pumps failed during the period from March 1962 through June 1962. Damage from these failures resulted in the loss of two MA-3 F-series Atlas missiles (missiles 1F and 11F), an explosion within MA-2 sustainer engine S/N 222080, and an explosion within Mark 4 turbopump S/N R112. Because of these failures, Rocketdyne initiated a program to reanalyze the Mark 4 design, manufacturing, assembly, inspection, and qualification techniques. No significant discrepancies were found during the analysis, but it was determined that some of the failures could be attributed to impeller and wear ring rubbing caused by deflection of the pump shaft. A proposal was submitted to resolve the problem by expending effort in the three main areas outlined below:

1. Develop a "fail-safe" LOX pump inlet adapter and wear ring. It was proposed to (a) fit a noncombustible Kel-F liner into the LOX pump inlet adapter, (b) delete the metal wear ring, and (c) fabricate the Kel-F liner in such a manner that it would also serve the function of the wear ring.
2. Conduct a components test program and an engine test program to determine the operating capability of the Kel-F liner.
3. The third proposed area of effort was divided into two parts: (a) to conduct a Mark 4 turbopump component test program to develop an instrumentation system which would measure LOX pump shaft deflections under actual operating conditions, and (b) to conduct an engine test program utilizing the MA-5 sustainer engine to determine the effect of various engine operating conditions on LOX pump shaft deflections.

The results of the Kel-F liner development program and the shaft deflection program are presented in this report.

SUMMARY

Four Mark 4 turbopumps failed during the first half of 1962, and Rocketdyne subsequently began a program to reanalyze the operation of the Mark 4 oxidizer pump. A review of the pump failures was also conducted, and it was positively determined that one of the failures was the result of an explosion caused when the oxidizer pump impeller rubbed against the wear ring. This condition was also strongly suspected as the cause of the other three pump failures.

To correct the problem, a "fail-safe" oxidizer pump inlet adapter was developed. The adapter was constructed with a Kel-F plastic liner, which is noncombustible under the conditions believed to have existed at the time of failure (pump rubbing in an atmosphere of liquid oxygen). An extensive test program totaling 225 MA-3 and MA-5 engine tests was conducted to test the device. A Mark 4 turbopump components test program involving 127 tests was conducted in support of the engine test program.

A portion of the testing was devoted to measuring the amount of deflection exhibited by the turbopump shaft under conditions such as normal main-stage operation, start transient, and conditions believed to be capable of causing malfunctions. A turbopump shaft deflection measuring device was designed and developed for this testing.

A study was also conducted to determine the factors which cause deflections that reduce the clearances within the oxidizer pump. These factors were found to be (1) external loads from the turbopump mounts and ducts, (2) internal hydrodynamic forces acting on the oxidizer impeller,

(3) internal hydrodynamic forces acting on the fuel impeller, (4) internal forces acting on the drive gear, and (5) the critical speed and unbalance of the shaft and associated masses.

Following are the major conclusions concerning the tests and analytical programs:

1. The Kel-F lined oxidizer pump inlet adapter is compatible with Atlas engine systems.
2. The possibility of Mark 4 oxidizer pump explosions due to rubbing between the inlet and the inducer or impeller has been eliminated by the incorporation of the "fail-safe" Kel-F liner.
3. No appreciable change in Atlas sustainer performance will be experienced as a result of incorporation of this modification, and no recalibration or adjustment of the engine will be required.
4. The amount of shaft deflection at the wear ring is as previously predicted; under the worst conditions, light rubbing might occur between the impeller and the Kel-F liner. A marginal condition existed with the metal wear ring and diverter lip.

PROGRAM BACKGROUND

The Mark 4 turbopump was designed for use in the sustainer engine of the Atlas missile. Development testing of this turbopump began in 1955. A number of failures within the LOX pump occurred between 1955 and 1959 while the turbopump and engine were undergoing development and change of sequencing, but no failures took place during a 27-month period from December 1959 to March 1962. From March through June 1962, four LOX pump failures occurred. These are discussed below.

The first Mark 4 turbopump failure occurred during March 1962 after approximately 21 seconds of the initial green run of MA-2 turbopump S/N R112. The oxidizer impeller, inducer, and wear ring were almost completely consumed in the ensuing fire, and the LOX primary seal also suffered the effects of some burning. The inlet adapter and pump volute were not cracked by high internal pressure, and the remainder of the pump was in relatively good condition. Inspection failed to reveal the cause of failure; from run data, it was postulated that the fire was caused by a rub that resulted from a structural failure of the oxidizer inducer or an impeller blade which was damaged or otherwise discrepant prior to the start of test. Other inducers and impellers produced during the same time period were reinspected, but no potential cause of failure was found.

The second failure occurred during April 1962 at 1.8 seconds after sustainer engine ignition start signal during the launch of Atlas missile 11F. Examination of the S/N R132R turbopump revealed probable initial rubbing between the impeller and the diverter lip. The volute was burst open by internal pressure, but was not burned. The inducer was dented, broken, and slightly burned, and the inlet adapter was badly burned. The inlet portion of the impeller and the diverter lip portion of the wear ring were burned. The remainder of the turbopump was essentially intact, but inspection revealed several discrepancies which probably resulted from the pump failure. Some of these discrepancies, if existent at the time of failure, could have caused or substantially contributed to the failure. Review of the data has not shown any engine system malfunction that would cause the LOX pump to rub, and the exact cause of this failure has not been determined.

The third Mark 4 turbopump failure occurred during May 1962 at 0.8 second after the MA-3 sustainer engine ignition start signal during a static test of Atlas missile 1F. Examination of the S/N R026R turbopump revealed probable initial rubbing between the impeller and diverter lip. The volute was burst open by internal pressure and other parts of the turbopump were extensively damaged. Examination of the test data revealed that the sustainer engine head suppression valve did not begin to open until 0.480 second after ignition of the solid-propellant gas generator. Subsequent testing with liquid nitrogen (Appendix D) showed that such a delay would cause the oxidizer pump to operate far from its design point and would cause the shaft to deflect an amount which would be more than sufficient to cause a rub. It was concluded that the pump failure was initiated when the delayed opening of the head suppression valve imposed excessive loading on the LOX pump. The most likely mode of failure of the valve was concluded to be contamination and/or moisture freezing in the shaft bearing.

The fourth Mark 4 pump failure occurred during June 1962 at 2.12 seconds after the ignition start signal during acceptance testing of an overhauled MA-2 engine. Examination of the S/N R055 turbopump revealed probable initial rubbing between the inducer and the inlet adapter. The inducer, the inlet adapter, the impeller, and the wear ring were burned extensively. The volute was not broken and was burned only slightly. The remainder of the turbopump was in good condition and gave no indication of the cause of the failure. Test data showed nothing that could be interpreted as abnormal prior to the initial burning. Because this was a recently overhauled turbopump, the overhaul technical procedures (Ref. 1), were examined. Although the overhaul technical order required revision with regard to measuring LOX pump clearances during assembly, the disassembly measurements disclosed no improper assembly conditions in the oxidizer pump and the inspection after turbopump green run had disclosed no rubbing. The cause of this failure has not been determined.

The four LOX pump failures in 1962 resulted in an extensive search for changes in design, procedures, and other items that may have been related to the failures. Although the change from hollow shaft to solid shaft was re-examined, the solid shaft is more rigid and failures occurred in turbopumps with both types of shafts. An investigation of manufacturing procedures revealed nothing that could contribute to the failures. It was found that some improvements could be incorporated into the assembly inspection procedures, and these changes have been put into effect at all locations. Several LOX pumps (including the LOX pump used in missile 1F) were reinspected, and no discrepancies were found. Test activities and facilities were studied without any significant findings. The Mark 4 turbopump design was reanalyzed and no deficiencies were noted.

Also precipitating from the four Mark 4 turbopump failures was a complete review of the history of the Atlas MA-2, MA-3, and MA-5 sustainer turbopumps. The purpose of this review was to determine the total number of turbopumps that have experienced discrepancies on the oxidizer side during green run.

The results of this review (Fig. 1 through 3) show that 17 of the 142 reviewed MA-2 sustainer turbopumps experienced rubbing problems, 4 of the 183 reviewed MA-3 sustainer turbopumps experienced rubbing, and 4 of the 52 reviewed MA-5 sustainer turbopumps experienced rubbing. All of the available four critical clearance dimensions (impeller axial clearance, diverter lip radial clearance, inducer-to-adapter clearance, and impeller-to-wear ring clearance) were recorded and averaged over the total turbopumps and over the turbopumps that had rubbing problems. The results are presented in Table 1. These dimensions showed no significant difference between turbopumps that rubbed and turbopumps that did not rub. The primary incidence of rubbing on the 17 MA-2 turbopumps occurred when the oxidizer impeller rubbed against the wear ring.

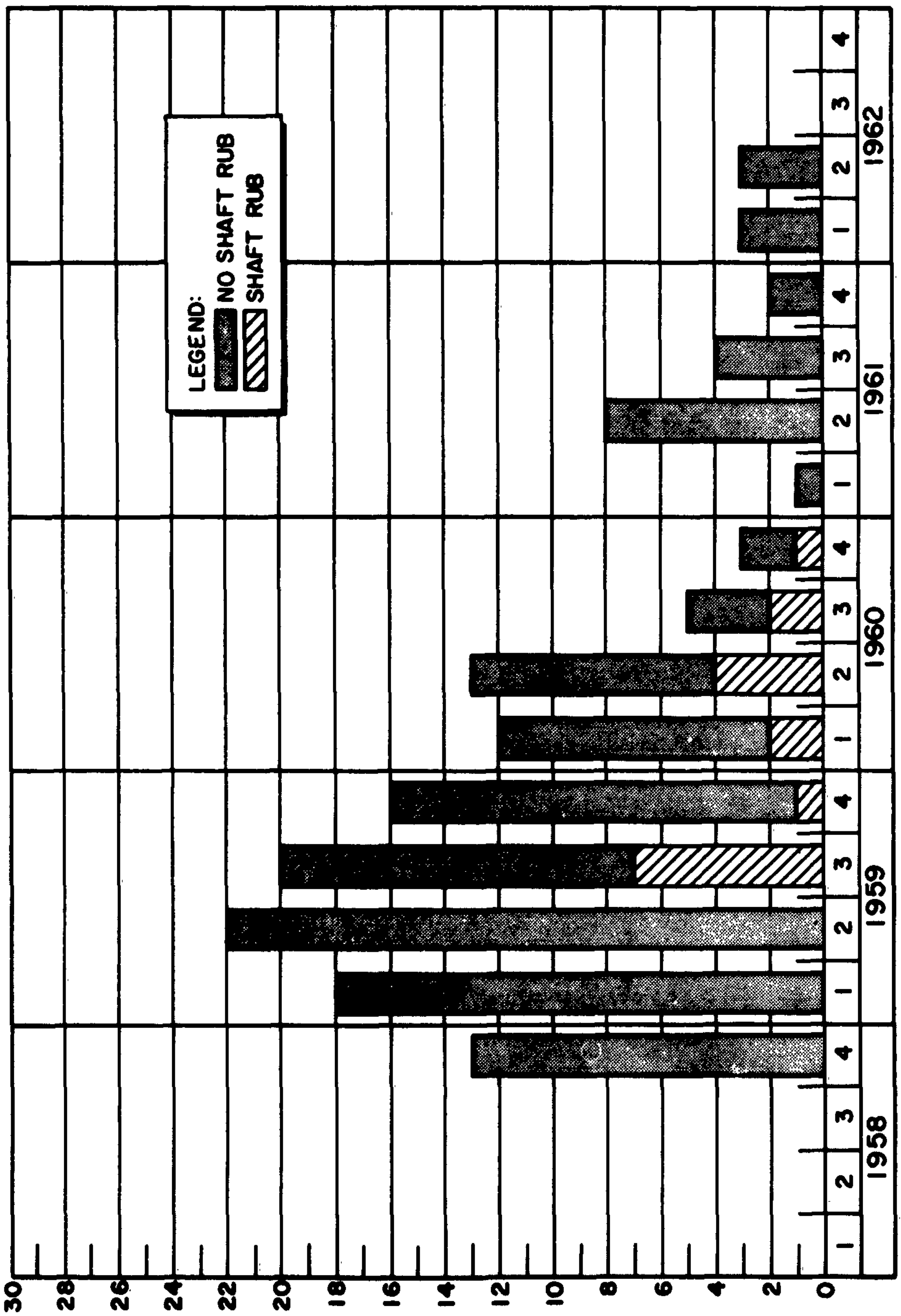

Figure 1. MA-2 Turbopump Green Run Rubbing History

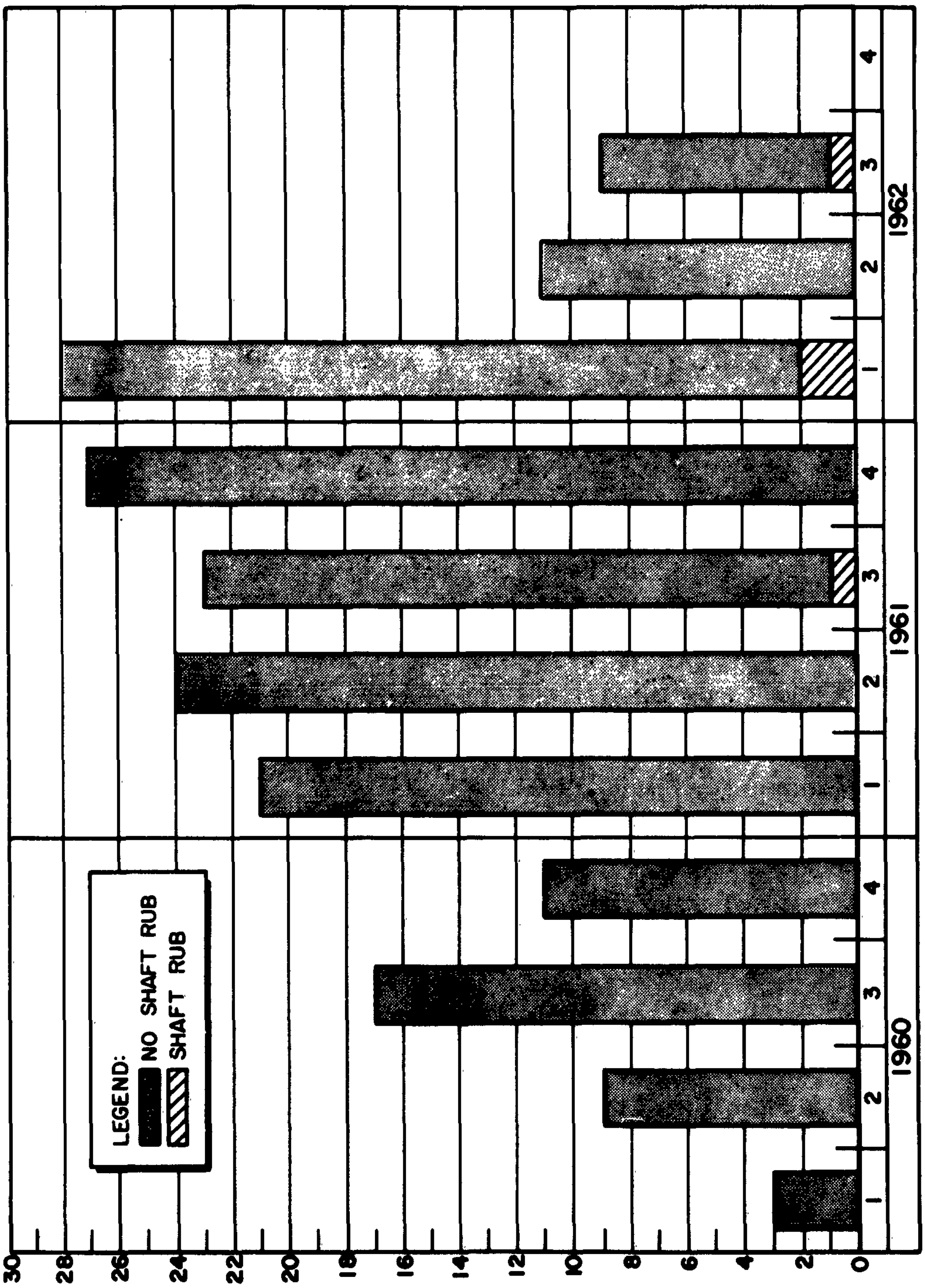

Figure 2. MA-3 Turbopump Green Run Rubbing History

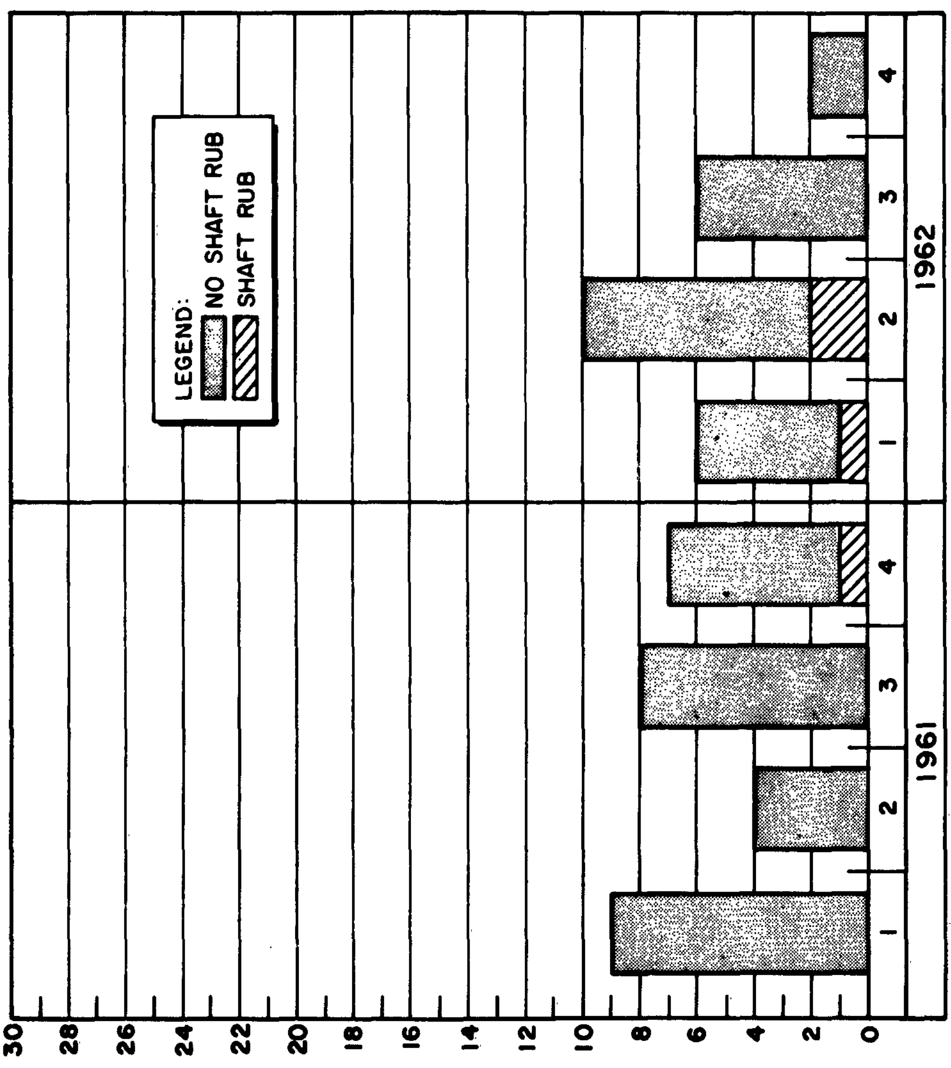

Figure 3. MA-5 Turbopump Green Run Rubbing History

TABLE 1

AVERAGE DIMENSIONS OF ATLAS MARK 4 TURBOPUMPS

Clearance Measurement	Total Turbopumps Tested			Turbopumps Experiencing Rubbing		
	Average Dimension, inch	High Dimension, inch	Low Dimension, inch	Average Dimension, inch	High Dimension, inch	Low Dimension, inch
MA-2 Impeller Axial	0.0719	0.078	0.065	0.0723	0.076	0.069
MA-2 Diverter Lip	0.041	0.045	0.040	0.0418	0.048	0.041
MA-2 Inducer-To-Adapter	0.0437	0.0495	0.040	0.0435	0.048	0.040
MA-2 Impeller-To-Wear Ring	0.032	0.036	0.028	0.0321	0.034	0.0295
MA-3 Impeller Axial	0.0722	0.075	0.065	0.0727	0.074	0.070
MA-3 Diverter Lip	0.0418	0.051	0.040	*	*	*
MA-3 Inducer-To-Adapter	0.0437	0.050	0.039	0.0431	0.045	0.041
MA-3 Impeller-To-Wear Ring	0.0317	0.036	0.025	0.0305	0.0345	0.0255
MA-5 Impeller Axial	0.0726	0.075	0.068	0.073	0.075	0.070
MA-5 Diverter Lip	0.0415	0.051	0.040	0.0416	0.045	0.040
MA-5 Inducer-To-Adapter	0.0437	0.049	0.039	0.041	0.043	0.039
MA-5 Impeller-To-Wear Ring	0.0318	0.053	0.0275	0.0306	0.033	0.029

*Dimensions verified to be within tolerance, but exact value not available

NOTE: Dimensions and Tolerances
- Impeller Axial Clearance, inch, 0.070 ±0.005
- Diverter Lip Clearance, inch, 0.040 minimum
- Inducer-To-Adapter Clearance, inch, 0.039 minimum
- Impeller-To-Wear Ring Clearance, inch, 0.025 minimum

ENGINE TESTS

To evaluate the operating capability of a Kel-F wear ring and LOX pump inlet adapter liner, 227 engine tests were conducted at the Rocketdyne Propulsion Field Laboratory. Atlas MA-5 and MA-3 sustainer engines were used, and testing was divided into five catagories: staging, limits, interference, performance, and endurance.

TEST DESCRIPTION

Staging Tests

The staging tests were conducted using a secondary prevalve in the facility LOX feed ducting. By partially closing the prevalve, a rapid drop in LOX pump inlet pressure could be induced to simulate the drop that occurs in flight at booster cutoff.

Limits Tests

During the limits tests, the engine was operated at combined thrust and mixture ratio extremes of ±7% thrust and ±15% mixture ratio.

As special limits investigations, tests were conducted in which turbopump overspeed conditions (11,200 rpm) were induced. In addition, tests were conducted where the engine was started with low available LOX net positive suction head (NPSH). One test was conducted with pretest engine environmental conditions to +130 F.

Interference Tests

The interference testing was performed using special inlets with reduced pilot diameters so that the inlet could be positioned to touch the impeller.

Performance Tests

Performance testing utilizes a special engine calibration which enables the head suppression valve to be full open without exceeding the nominal engine mixture ratio +15%. From a nominal thrust and mixture ratio operating point, the propellant utilization valve angle is reduced in 2-degree increments until the head suppression valve goes to the full-open position (out of control). The propellant utilization valve is then driven to the mechanical stop, and the engine is operated in this orificed configuration to obtain stabilized data. Two performance tests with a standard all-metal wear ring were conducted on each engine tested, and were followed by two performance tests with a Kel-F-lined inlet. Engine performance shifts resulting from retrofit of the Kel-F inlet were determined by comparing the data obtained from these tests.

Endurance Tests

The endurance testing subjected the liner to normal engine acceptance test conditions. The engine was operated at nominal thrust and mixture ratio, with excursions to the ±15% mixture ratio extremes. In general, all other testing contributed to the accumulation of endurance time on Kel-F inlets.

MA-3 ENGINE TESTS

The purpose of the Kel-F liner evaluation with the MA-3 sustainer engine was to: (1) demonstrate the effects caused by liner-to-impeller contact,

(2) expose the liner to all conceivable engine operating conditions (low NPSH start, high and low thrust accompanied by high and low mixture ratio, and simulated booster cutoff or staging), and (3) demonstrate liner durability.

The MA-3 engine test program consisted of 83 engine tests totaling 15,052 seconds of test time (Table 2) with 9 Kel-F liners. Two additional tests were conducted with a metal wear ring.

Posttest inspection revealed that all 9 Kel-F liners assigned to the MA-3 test series of 83 engine tests were in completely satisfactory condition. A complete summary of these tests is presented in Table 3.

Interference Tests

Twenty tests were conducted with interference deliberately established between the impeller and the liner. Interference was accomplished by subjecting the engine to unusual pump-loading conditions which resulted in abnormal shaft deflection. During 17 tests, the loading was created by not opening the head suppression and propellant utilization valves while firing the solid-propellant gas generator with primed pumps. During two tests, the head suppression valve delay experienced with missile 1F was simulated. This delay was approximately 480 milliseconds from ignition start. The final interference test was conducted with a normal engine start sequence. Mainstage duration was accomplished.

The amount of interference was determined by measuring the depth of wear on the Kel-F liner. Interferences were recorded from 0.001 to 0.009 inch, with the majority in the range of 0.004 to 0.007 inch. To maximize the interference, the pilot diameter of the inlet adapter was reduced and the adapter was installed with a clearance of 0 to 0.001 inch at the expected interference location.

TABLE 2

MA-3 ENGINE TEST TIME ACCUMULATED ON KEL-F LINERS

Liner Serial No.	Number of Tests	Total Accumulated Test Time, seconds
2	19	4553
3	8	904
3-1	10	1502
5	9	1550
6	1	0*
6-1	7	900
6-2	7	900
7	8	1883
8	14	3860
Standard Metal	2	300

*One ignition-stage only test

Three liners experienced four rubs each (one rub in each quadrant of the liner); one liner experienced five rubs (one on top of another); one liner was rubbed once; and one liner was rubbed twice. Each liner experienced one test conducted with a starting NPSH of 30 feet. The minimum recommended starting NPSH for the MA-3 sustainer engine is 40 feet. Test results revealed no evidence that particles of Kel-F rubbed from the liner had a harmful effect on the engine. Additionally, there was no evidence that any liner was damaged severely enough to cause uncertainty in its integrity.

Staging Tests

Four tests were conducted to investigate the effect of a sudden decrease in LOX pump inlet pressure. It was surmised that the pressure in the cavity on the outside diameter of the liner might not vent as rapidly as the pressure decrease inside the liner, causing the liner to collapse. The missile event causing this situation is booster cutoff, at which time the sustainer LOX pump inlet pressure drops from approximately 120 to 20 psi within 300 milliseconds. This sequence was simulated by the installation of the previously mentioned 6-inch auxiliary prevalve in the sustainer low-pressure LOX ducting. The valve actuator was fitted with a spool that limited its travel and, therefore, valve closure.

The first test resulted in a pressure drop of 45 psi. The next three tests resulted in pressure drops of 70, 110, and 140 psi. Three liners were used during these tests, and the 70- and 110-psi pressure-drop tests were conducted with the same liner. Each test consisted of three pressure drops, but the effect of these conditions on the liners was undetectable. One liner subsequently accumulated 1500 seconds during five tests; this liner previously had accumulated 2060 seconds during eight tests. During one of the tests, the sustainer engine was operated at approximately 40,000 pounds of thrust for 50 seconds after reduction of LOX pump inlet NPSH to 9 feet.

Limits Tests

Ten tests totaling 3000 seconds were conducted as limits tests. These tests were made with thrust levels of ±7% of nominal and mixture ratio excursions of ±15% of nominal. No adverse effects were noted during these tests.

No adverse effects were noted upon posttest inspection of the liner after one 300-second test preceded by engine environmental conditioning to +130 F. Engine operation and performance during the test were satisfactory.

Pressure-Instrumented Tests

In conjunction with the engine tests, pressure measurements were made at six locations on the LOX pump volute. Four pressure conditions are shown in Fig. 4, but the pressure values do not represent a highly accurate pressure profile of the impeller because the pressure transducer location varies in distance from the impeller. The conditions represent steady-state situations where there is essentially no pressure transient (less than 0.3 psi/msec).

Condition 1 is a representative mainstage condition and, as expected, exhibits essentially no differential pressure around the volute.

Condition 2 is representative of the maximum pressure variation during a normal start sequence. The time in the sequence was 85 milliseconds after main propellant ignition. At this time, both main valves were wide open, bootstrap had not yet occurred, and the differential pressure from position 2 to position 6 was 40 psi.

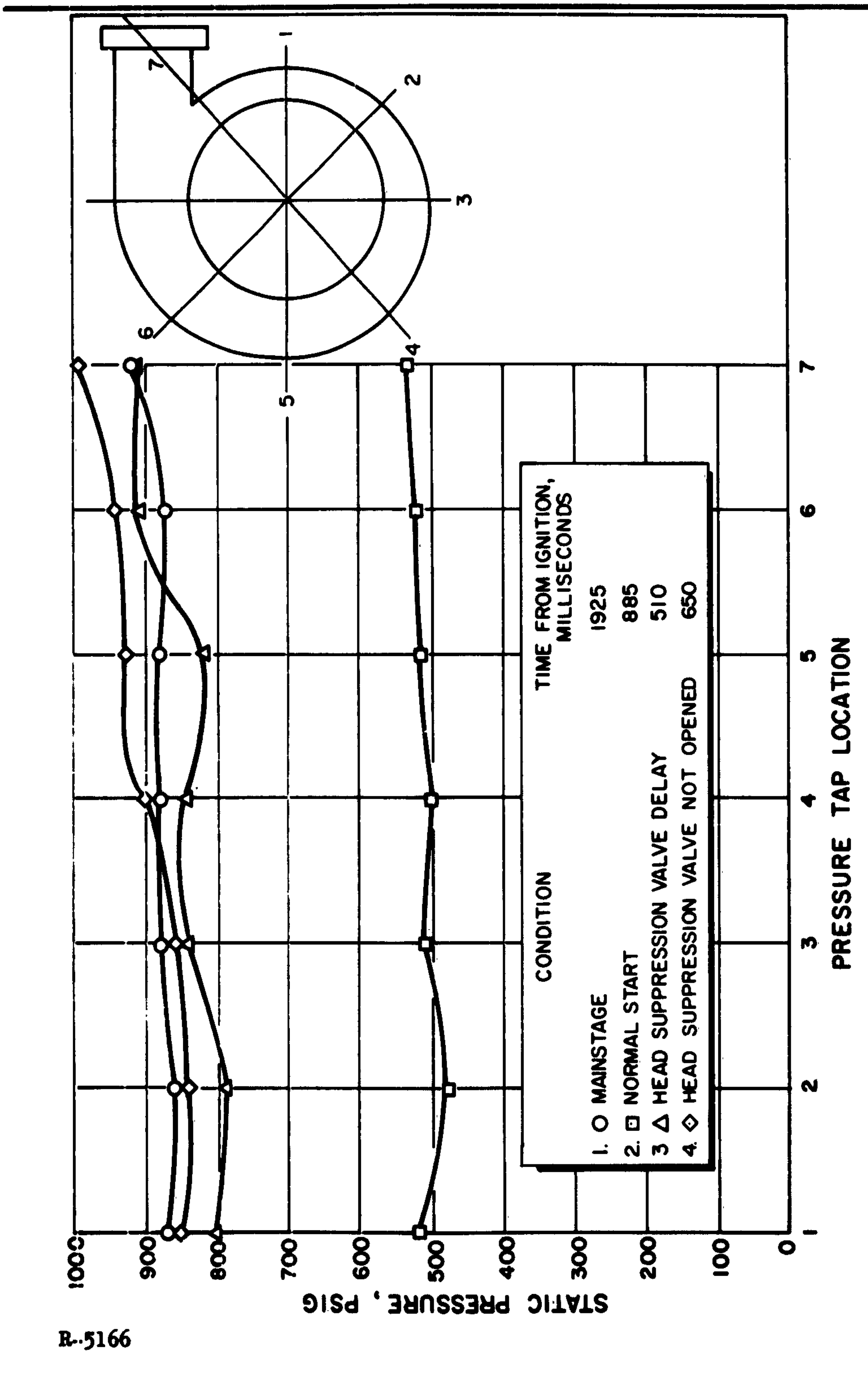

Figure 4. Pressure-Instrumented Volute Tests

Condition 3 represents a time period 35 milliseconds after the head suppression valve started to open, following a 475-millisecond opening delay. The time is just prior to the pressure decay caused by the opening head suppression valve. The differential pressure from position 2 to position 6 at that instant was 120 psi.

Condition 4 is representative of the pressure profile when the head suppression valve is not opened. The maximum differential pressure from position 2 to position 6 was 100 psi.

It was hypothesized that if conditions 3 and 4 existed, the pressure differentials around the pump volute would be great enough to cause shaft deflection and rubbing. The tests seem to validate this hypothesis.

TABLE 3

TEST RESULTS, LINER INSTALLED IN MA-3 ENGINE

Test No.	Liner Serial No.	Test Duration, seconds	Accumulated Duration, seconds	Comments
535-191	5	0*	0	Bootstrap failure
-192	5	50	50	
-193**	5	0*	50	Head suppression valve not opened
-194	5	300	350	
-195**	5	0*	350	Head suppression valve not opened
-197	5	300	650	
-198	5	300	950	-7% thrust; ±15% mixture ratio
-199	5	300	1250	+7% thrust; ±15% mixture ratio
-200	5	300	1550	30-foot starting NPSH***
535-196**	6	0*	0	Head suppression valve not opened
535-201**	6-1	0*	0	Head suppression valve not opened
-202	6-1	300	300	-7% thrust; ±15% mixture ratio
-203**	6-1	0*	300	Head suppression valve not opened
-204	6-1	300	600	+7% thrust; ±15% mixture ratio
-205**	6-1	0*	600	Head suppression valve not opened
-206	6-1	300	900	30-foot starting NPSH
-207**	6-1	0*	900	Head suppression valve not opened

*Solid-propellant gas generator only test
**Impeller was rubbed against Kel-F liner
***NPSH = net positive suction head

TABLE 3

(Continued)

Test No	Liner Serial No	Test Duration, seconds	Accumulated Duration, seconds	Comments
535-208*	3	0**	0	Head suppression valve not opened
-209	3	300	300	-7% thrust, ±15% mixture ratio
-210*	3	0**	300	Head suppression valve not opened
-211*	3	4	304	
-212	3	300	604	+7% thrust, ±15% mixture ratio
-213*	3	0**	604	Head suppression valve not opened
-214*	3	0**	604	Head suppression valve not opened
-215	3	300	904	30-foot starting NPSH**
535-216*	6-2	0**	0	Head suppression valve not opened
-217	6-2	300	300	-7% thrust; ±15% mixture ratio
-218*	6-2	0**	300	Head suppression valve not opened
-219	6-2	300	600	+7% thrust; ±15% mixture ratio
-220*	6-2	0**	600	Head suppression valve not opened
-221	6-2	300	900	30-foot starting NPSH
-222*	6-2	0**	900	Bootstrap failure; 480-millisecond head suppression valve delay (simulated missile 1F)

*Impeller was rubbed against Kel-F liner

**Solid-propellant gas generator only test

TABLE 3

(Continued)

Test No.	Liner Serial No.	Test Duration, seconds	Accumulated Duration, seconds	Comments
535-227*	3-1	0**	0	Head suppression valve not opened
-228	3-1	300	300	-7% thrust; ±15% mixture ratio
-229*	3-1	0**	300	Head suppression valve not opened
-230	3-1	2**	302	
-231	3-1	300	602	+7% thrust; ±15% mixture ratio
-232*	3-1	0**	602	Head suppression valve not opened
-233	3-1	300	902	30-foot starting NPSH
-234*	3-1	0**	902	Bootstrap failure; 480-millisecond head suppression valve delay (simulated missile 1F)
-235	3-1	300	1202	
-236	3-1	300	1502	Pretest environmental conditioning at +130 F
513-020	2	50	50	
-021	2	280	330	
514-069	2	200	530	
-070	2	280	810	
-071	2	280	1090	
-072	2	280	1370	

*Impeller was rubbed against Kel-F liner
**Solid-propellant gas generator only test

TABLE 3

(Continued)

Test No.	Liner Serial No.	Test Duration, seconds	Accumulated Duration, seconds	Comments
513-022	2	280	1650	
-023	2	50	1700	
-024	2	280	1980	
-025	2	280	2260	
-026	2	280	2540	
-027	2	280	2820	
-028	2	273	3093	
-029	2	280	3373	
-030	2	280	3653	
525-223	Standard	150	—	
-224	2	150	3803	
-225	Standard	150	—	
-226	2	150	3953	
-238	2	300	4253	Simulated staging at 70 psi
-239	2	300	4553	Simulated staging at 110 psi
513-031	8	260	260	
-032	8	280	540	
-033	8	280	820	
-034	8	280	1100	
-035	8	300	1400	
-036	8	100	1500	
-037	8	280	1780	
-038	8	280	2060	

TABLE 3
(Continued)

Test No.	Liner Serial No.	Test Duration, seconds	Accumulated Duration, seconds	Comments
535-240	8	300	2360	Simulated staging at 140 psi
-241	8	300	2660	
-242	8	300	2960	
-243	8	300	3260	
-244	8	300	3560	
-245	8	300	3860	
513-039	7	280	280	
-040	7	280	560	
-041	7	3	563	Premature cutoff
-042	7	280	843	
-043	7	280	1123	
-044	7	280	1403	
-045	7	280	1683	
535-237	7	200	1883	Simulated staging at 44 psi
535-223	Standard	150	--	
535-225	Standard	150	--	

MA-5 ENGINE TESTS

Six Atlas MA-5 sustainer engines were subjected to a total of 142 tests during the program, and the total time accumulated during these tests was 18,78 seconds (Table 4). The engines were subjected to four different types of testing: (1) performance, (2) endurance, (3) interference, and (4) limits.

Performance Testing

Twenty-two performance tests were conducted, accumulating 4748 seconds of engine test time. Seven of the tests were conducted with standard all-metal LOX pump inlet adapters and wear rings; these tests involved 1568 seconds of test time. Fifteen tests totaling 3180 seconds of test time were conducted with Kel-F-lined LOX pump inlet adapters; the purpose of this testing was to evaluate the change in LOX pump performance, if any, and any associated change in engine performance resulting from retrofitting an engine with a Kel-F liner without reacceptance testing or otherwise recalibrating that engine.

Data obtained from performance testing on engine S/N 022-4 (tests 513-048, 513-049, 513-056 and 513-058) were later disqualified for performance considerations for two reasons: (1) only one performance test was conducted with the standard all-metal inlet adapter and wear ring, and (2) subsequent inspection of the metal wear ring after removal revealed evidence of rubbing.

The valid performance tests were compared on the basis of the change in developed heat at a given volumetric flowrate (1204 gpm) and pump speed (10,161 rpm). A direct comparison of the change in LOX pump and engine performance caused by the Kel-F liner was unreliable because of other

TABLE 4

TEST RESULTS, LINER INSTALLED IN MA-5 ENGINE

Test No.	Liner Serial No.	Test Duration, seconds	Accumulated Duration, seconds	Comments
513-046	Standard	2	--	Overspeed; vernier engine bleeds did not open
-047	Standard	52	--	
-048	Standard	191	--	Performance test
-049	9X	57	57	Performance test
-050	9X	255	312	
-051	9X	250	562	
-052	9X	250	812	
-053	9X	300	1112	
-054	9X	297	1409	
-055	9X	300	1709	
-056	12	231	231	Performance test
-057	12	100	331	
-058	12	158	489	Performance test
-059	12	100	589	
-060	12	72	661	
-061	12	70	731	
-062	12	70	801	
-063	12	70	871	
-064	12	70	941	
-065	12	71	1012	

TABLE 4

(Continued)

Test No.	Liner Serial No.	Test Duration, seconds	Accumulated Duration, seconds	Comments
513-066	Standard	240	--	Performance test
-067	Standard	241	--	Performance test
-068	9X	241	1950	Performance test
-069	9X	242	2192	Performance test
-070	9X	2	2194	Overspeed; vernier bleed purposely capped
-071	4X	241	241	Performance test
-072	4X	251	492	Performance test
-073	Standard	255	--	Performance test
-074	Standard	240	--	Performance test
-075	5-1	241	241	Performance test
-076	5-1	242	483	Performance test
-077	5-1	255	738	
-078	5-1	261	999	
-079	5-1	261	1260	
-080	5-1	261	1521	
-081	5-1	292	1813	
-082	4X	66	558	
-083	4X	66	624	
-084	4X	66	690	
-085	4X	65	755	

TABLE 4
(Continued)

Test No.	Liner Serial No.	Test Duration, seconds	Accumulated Duration, seconds	Comments
513-086	4X	66	820	
-087	4X	66	886	
-088	4X	65	951	
-089	4X	66	1017	
-090	13	290	290	
-091	13	290	580	
-092	13	291	871	
-093	13	291	1162	
-094	13	290	1452	
-095	13	176	1628	Propellant utilization valve excursion
-096	13	175	1803	Propellant utilization valve excursion
-097	14	202	202	Performance test
-098	14	221	423	Performance test
-099	14	76	499	
-100	14	76	575	
-101	14	76	651	
-102	14	76	727	
-103	14	76	803	
-104	14	76	879	
-105	14	76	955	
-106	14	76	1031	

TABLE 4
(Continued)

Test No.	Liner Serial No.	Test Duration, seconds	Accumulated Duration, seconds	Comments
513-107	16	100	100	Propellant utilization valve excursion
-108	16	101	201	Propellant utilization valve excursion
-109	16	101	302	Propellant utilization valve excursion
-110	16	100	402	
-111	16	100	502	
-112	16	100	602	
-113	16	100	702	
-114	16	100	802	
-115	16	101	903	
-116	16	101	1004	
-117	15	100	100	
-118	15	100	200	
-119	15	102	302	
-120	15	101	403	
-121	15	101	504	
-122	15	100	604	
-123	15	101	705	
-124	15	100	805	
-125	15	100	905	
-126	15	100	1005	

TABLE 4
(Continued)

Test No.	Liner Serial No.	Test Duration, seconds	Accumulated Duration, seconds	Comments
513-127	17	101	101	Propellant utilization valve excursion
-128	17	100	201	Propellant utilization valve excursion
-129	17	101	302	Propellant utilization valve excursion
-130	17	102	404	
-131	17	51	455	
-132	17	51	506	
-133	17	50	556	
-134	Standard	201	--	Performance test
-135	Standard	200	--	Performance test
-136	17	200	756	Performance test
-137	17	251	1007	Performance test
-138	10X	201	201	Performance test
-139	10X	201	401	Performance test
-140	10X	76	477	
-141	10X	75	552	
-142	10X	75	627	
-143	10X	75	702	
-144	10X	76	778	
-145	10X	76	854	
-146	10X	75	929	
-147	10X	75	1006	

TABLE 4
(Continued)

Test No.	Liner Serial No.	Test Duration, seconds	Accumulated Duration, seconds	Comments
513-148	18	100	100	Propellant utilization valve excursion
-149	18	101	201	Propellant utilization valve excursion
-150	18	100	301	Propellant utilization valve excursion
-151	18	100	401	
-152	18	100	501	
-153	18	100	601	
-154	18	100	701	
-155	18	101	802	
-156	18	100	902	
-157	18	100	1002	
-158	19	101	101	
-159	19	101	202	
-160	19	100	302	
-161	19	100	402	
-162	19	100	502	
-163	19	101	603	
512-123	19	19	622	
-124	19	0	622	
-125	19	100	722	
-126	19	10	732	
-127	19	270	1002	

TABLE 4
(Continued)

Test No.	Liner Serial No.	Test Duration, seconds	Accumulated Duration, seconds	Comments
512-129	20	270	270	
-130	20	35	305	
-131	20	270	575	
-132	20	127	702	Overspeed; orifice in wrong location
-133	20	270	972	
-134	20	270	1242	
-135	20	270	1512	
-136	20	271	1783	
-137	9X	10	2204	Shaft deflection
-138	9X	10	2214	Shaft deflection
513-001	9X	70	2284	Shaft deflection
-002	9X	70	2354	
-003	9X	200	2554	Fixed propellant utilization system
-004	9X	200	2754	Fixed propellant utilization system
-005	9X	200	2954	Fixed propellant utilization system; MA-2 mixture ratio control
-006	9X	200	3154	Fixed propellant utilization system; MA-2 mixture ratio control
-007	6-3	10	10	Offset inlet; fixed propellant utilization system; MA-2 mixture ratio control
-008	6-3	10	20	Offset inlet

uncontrolled variables. Therefore, LOX pump performance data were isolated and reduced to the above-quoted common conditions to allow comparison. The change in developed head between the standard all-metal inlet adapter and wear ring and the Kel-F-lined inlet adapter was calculated for each liner tested. The mean change in developed head was calculated to be an increase of 12 feet, based on three turbopumps and six Kel-F-lined inlet adapters.

Component tests described later in this report were conducted with 10 pumps to determine the performance shift caused by installation of the Kel-F liner. These tests showed that the mean increase in developed head was 26 feet; this figure is considered more reliable because of the larger sample and fewer uncontrolled variables affecting test-to-test repeatibility. Using the figure of 26 feet, the effect of Kel-F-liner retrofit on nominal sustainer engine rated performance was calculated and is summarized in Table 5. The change in performance is considered insignificant.

Endurance Testing

In general, all testing contributed to the accumulation of endurance time on Kel-F-lined LOX pump inlet adapters, but MA-5 test time was accumulated for the specific purpose of demonstrating Kel-F-liner endurance. Testing for endurance involved 106 MA-5 sustainer engine tests, and the accumulated test time was 13,497 seconds. Thirteen Kel-F-lined LOX pump inlet adapters were tested during this phase of the test series.

The propellant utilization valve angle was varied during 11 of the tests so that ±15% mixture ratio extremes could be produced. The +15% mixture ratio corresponds to an oxidizer-to-fuel mixture ratio of approximately

TABLE 5

EFFECT OF KEL-F LINER RETROFIT ON
NOMINAL SUSTAINER ENGINE RATED PERFORMANCE

Parameter	Standard Inlet	Kel-F Liner	Difference
Thrust, pounds	57,000	57,000	0
Mixture Ratio, oxidizer/fuel	2.270	2.270	0
Specific Impulse, seconds	214.5	214.5	+0.1
Pump Speed, rpm	10161	10161	0
LOX Pump Head, feet	1889	1915	+26
LOX Flow, gpm	1204	1204	0
Turbine Power, bhp	1688	1702	+14
Turbine Inlet Temperature, F	1059	1068	+9
Head Suppression Valve Differential Pressure, psi	162.8	175.9	+13.1

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2.62, while a -15% mixture ratio corresponds to an approximate 1.93 mixture ratio. At a +15% mixture ratio, the nominal 57,000 pounds of MA-5 sustainer thrust is raised to 57,300 pounds (an increase of approximately 0.52%), and the nominal 215-second specific impulse is lowered to approximately 214.5 seconds. At -15% mixture ratio, the nominal thrust level is lowered to approximately 55,700 pounds (a decrease of 2.28%), and the nominal specific impulse is decreased to approximately 210 seconds. Therefore, operation of the MA-5 sustainer engine in the -15% mixture ratio range has a more pronounced effect upon engine performance than operation in the +15% mixture ratio range.

Upon completion of testing, each liner was removed and inspected for rubbing, cracking, distortion, or any other undesirable condition. None of the 14 liners tested was found to have conditions which could be considered degrading to operational or endurance capability.

The liners tested during the MA-5 test series and the total test time which each liner accumulated is summarized below:

Liner Serial No.	Total Accumulated Test Time, seconds
4-X	1017
5-1	1813
6-3	961
9-X	2214
10-X	1006
12	1012
13	1803
14	1031
15	1005
16	903
17	1007
18	1002
19	1002
20	1783

Interference Testing

Two interference tests were conducted, and rubbing between the LOX pump impeller and the Kel-F liner was purposely induced during the tests. Twenty seconds of test time were accumulated and LOX pump shaft deflection was measured during each test.

One test was conducted with a fixed propellant utilization system (to increase the propellant utilization valve control rate), an MA-2 mixture ratio controller (to increase the head suppression valve control rate), and an offset LOX pump inlet adapter (to induce rubbing). The second test was conducted with standard MA-5 hardware, except that the Kel-F-lined LOX pump inlet adapter was offset to induce rubbing between the liner and the impeller.

During the first test, the engine was operated for 10 seconds with a normal start and cutoff sequence. At the conclusion of the test the LOX pump inlet adapter was removed and inspected. Some rubbing was found, but this rubbing had occurred during a component test with this liner. The liner was reinstalled on the engine and another 10-second test was conducted. A normal start and cutoff sequence was used during the second test, and the LOX pump inlet adapter was removed and examined at the conclusion of the test. New rub marks were found in the 10 to 2 o'clock region, and one small strand of Kel-F remained attached to the main body of Kel-F. Examination of the LOX pump, associated hardware, and the test data revealed that engine operation and performance had not been adversely affected as a result of the impeller rubbing against the Kel-F liner.

Limits Test

One test in the MA-5 series was conducted as a special limits test. During this test, the vernier engine simulation bleeds were purposely capped off so the turbine would experience an overspeed condition (approximately 11,200 rpm). The overspeed condition existed because the LOX regulator cannot close sufficiently to compensate for a decreased flow demand resulting from capping of the simulation bleeds. This causes the gas generator LOX flowrate to exceed normal operating limits. A corresponding increase occurs in gas generator mixture ratio, combustion temperature, power output, and turbine speed. The test was terminated by a turbine overspeed trip device when the turbine operating limit was reached. The test duration was 2.7 seconds from ignition start (or 2.1 seconds from flight lockin signal).

The purpose of the limits test was to determine if turbine overspeed conditions would cause the inducer or the impeller to rub against the Kel-F liner. The Kel-F liner and inlet adapter were not offset to induce rubbing, but were installed in accordance with normal operational procedures. The liner was removed and inspected following the test, but no trace of rubbing was found.

Because of an orificing error, another turbine overspeed occurred during test 512-132. No adverse effects were noted.

COMPONENT TESTS

ELEMENTAL TESTING

A portion of the development program was devoted to special tests to substantiate the validity of the thermal analysis and specific features incorporated into a Kel-F-Lined LOX pump inlet adapter. These tests are discussed in this section.

Storage Capabilities

The requirement that the turbopump be capable of storage at temperatures up to 160 F necessitated the determination of whether Kel-F is susceptible to aging which is accelerated at elevated temperatures, and whether Kel-F would exhibit excessive creep at elevated temperatures. Excessive creep is undesirable because it can result in loose bolts and loose fits.

Aging. Concerning the aging effects, analysis has shown no chemical degradation of Kel-F with time. Kel-F material several years old exhibits the same mechanical and chemical properties as new material, so storing Kel-F-lined inlets for extended durations is not limited by aging effects.

Mechanical Effect of Elevated Temperatures. The model specification requires that turbopump components function after exposure to 160 F temperatures, so a press-fit Kel-F-lined inlet was subjected to this temperature for a continuous 24-hour period.

The inlet was returned to room temperature and subsequent measurements revealed that the press fit had been transformed to a slip fit, indicating that some plastic flow did occur during the 24-hour "heat soak." This loosening of the liner within the housing is not detrimental to the function of the inlet, but the manufacturing process was altered to include the heat soak as part of the assembly procedure. The process further requires that final bolt torquing and machining be performed after completion of the heat soak. This allows proper running clearances to be obtained between the liner, the impeller, and the inducer. This manufacturing procedure was reflected in the turbopump configuration used throughout the development test program.

To better define the creep characteristics of Kel-F, a test program was initiated utilizing Kel-F samples heated to specified temperatures and then subjected to various initial compressive loads. The elevated temperature (160 F) was maintained, and the load decay caused by creep was recorded at various intervals of time. Results of the testing, summarized in Appendix A, show that the percentage decrease in stress for a specified time is essentially the same for a given Kel-F sample regardless of the initial loading. Samples tested at initial stresses of 300 to 2200 psi after 24 hours of heating at 160 F showed a reduction in stress to approximately 57% of the original value. After 50 hours at 160 F, the stress is reduced to about 50% of the original value. One test involved heating a Kel-F sample at 160 F for 24 hours, retorquing the sample to its original stress level of 350 psi, and reheating for an additional 1375 hours. At the end of this time, the stress had not yet decreased to the low value attained prior to retorquing after only 24 hours. From this test it is evident that the procedure of retightening the bolts after the 24-hour heat soak significantly assists in stabilizing the Kel-F creep for the life of the part.

The thickness-to-diameter (t/d) ratio of the Kel-F sample also affected the creep or relaxation process at elevated temperatures. Under identical environmental conditions, samples with small t/d ratios showed less creep than samples with large t/d ratios. As was anticipated, the magnitude of Kel-F relaxation increased as temperature increased.

Bolt Lock

The bolts retaining the support ring and liner are prevented from rotating by using locking inserts in the housing. To evaluate the effectiveness of the locking insert, a number of inlets were checked to establish the locking torque. Although the average lock torque was between 3 to 4 in.-lb, a few inserts were found to have locking torques below 1 in.-lb. To protect against using inserts with low locking torques, the drawings were revised to specify a minimum acceptable torque callout of 2 in.-lb.

Six inlets were also checked after testing to determine the possibility of bolts loosening during the tests. The lowest bolt loosening torque recorded was 11 in.-lb, and all other bolt torques were 15 in.-lb or better.

As a result of the analysis and testing conducted, the bolt lock is considered (1) safe and (2) to have demonstrated adequate reliability.

Invar-36 Material

Particular attention was devoted to the Invar material because of the limited reliable information available with respect to thermal coefficients.

Also, there was some evidence that the thermal coefficient was related to the material hardness, and that varying degrees of cold-working would produce a marked change in the thermal contraction characteristics. To resolve these questions, an Invar specimen was fabricated from the same batch of material used to manufacture the development parts. The specimen was tested to determine the thermal contraction characteristics, then was annealed and the thermal characteristics were redetermined. The results of this test verified that the Invar used for all spacers throughout the development program did not differ from annealed Invar, so all future spacers will be machined from annealed material for uniformity and the drawing changed to reflect the annealing process. The investigation evaluating the thermal characteristics of Invar showed that, between +170 to -300 F, the average coefficient of thermal contraction of Invar-36 is $1.3 \pm 0.2 \times 10^{-6}$ in./in. F.

TURBOPUMP TESTING

To substantiate the design of the Kel-F-lined LOX pump inlet, a development program was initiated to evaluate significant areas in which the inlet is required to demonstrate satisfactory operation. These areas of operation encompass the design requirements and include (1) performance, (2) reliability and (3) fail-safe capabilities.

A total of 41,697 seconds of turbopump and LOX pump testing was accomplished during this development program. The test program utilized 12 different inlets and four turbopumps. Four of the inlets tested each accumulated in excess of 6200 seconds of operational time. Throughout all the testing (pump, turbopump, and engine) no difficulties were encountered which could be attributed to the Kel-F inlet. Tables 6 and 7 summarize the testing performed on the turbopump and LOX pump.

TABLE 6

LOX PUMP TEST SUMMARY, KEL-F-LINED LOX PUMP INLET

Inlet Serial No.	Test Time, Seconds	No. of Tests	Test Description
1	763	12	Performance tests, standard installation, lined inlet
2	464	6	
3	437	6	
4	474	6	
5	509	6	
6	522	6	
7	522	6	
8	466	6	
9	520	6	
10	441	6	
11	1125	12	Axial clearance tests
12	512	6	Rubbed Kel-F-lined inlet

TABLE 7

TURBOPUMP TEST SUMMARY, KEL-F-LINED L

Turbopump Serial No.	Inlet Serial No.	Endurance Tests: Test Time, seconds	Endurance Tests: No. of Tests	Cavitation Tests: Test Time, seconds	Cavitation Tests: No. of Tests	Pressure Tests: Test Time, seconds	Pressure Tests: No. of Tests
R082R and R015R	1	6383	21				
R015R	2	4305	19				
R015R	3-1	..				*655	5
R082R and R005R	4	6000	20	363	7		
R082R and R005R	5	6333	23				
R082R	6	..					
R015R	6-2	20	2				
R015R	6-3	..					
R015R	7	1874	20				
R015R and R016R	8X	6231	22				
TOTAL		31,146	127	363	7	*655	5

*Tests performed concurrently.

TABLE 7

UMMARY, KEL-F-LINED LOX PUMP INLET

Pressure Tests		Interference Tests		Limits Tests		Total Tests	
Test Time, seconds	No. of Tests	Test Time, seconds	No. of Tests	Test Time, seconds	No. of Tests	Test Time, seconds	No. of Tests
				600	.2	6983	23
						4305	19
*655	5	*28	2			655	5
				600	2	6963	29
						6333	23
		28	3			28	3
		20	2			40	4
		30	2			30	2
				600	2	2474	22
				900	3	7131	25
*655	5	78 *28	7 2	2700	9	34,942	155

2

The following paragraphs describe the tests and results obtained during the development phase of the program.

Endurance Tests

The purpose of the endurance testing was to demonstrate mechanical integrity of the inlet design and show that the inlet design could perform under normal operating conditions for the anticipated overhaul life of 6100 seconds. The testing consisted of operating the turbopump at nominal acceptance test conditions with the LOX pump settings targeted to the following values:

Pump Speed, rpm	10,000
Pump Flow, gpm	1180
Pump Discharge Pressure, psig	980
Inlet Pressure, psig	50

Endurance runs were scheduled for a duration of 300 seconds, which is the normal operating time for the Mark 4 turbopump. A total of 31,146 seconds of turbopump testing was accumulated on seven Kel-F-lined inlets subjected to the LOX pump conditions outlined above. No difficulties were experienced, and three of the inlets were tested in excess of 6200 seconds each during this phase of testing.

Examination of the liners after testing revealed no perceptible changes. Two inlet assemblies were disassembled after 9630 and 7860 seconds of accumulated operation, and were reinspected in accordance with the production drawing. The accumulated operating time included pump calibration, turbopump testing, and engine testing.

After inspection, the parts were reassembled and reidentified to the production drawing number. Each part was acceptable for continued use.

Interference (Rubbing) Tests

The interference testing was performed to demonstrate that no detrimental effects would occur if the impeller rubbed against the Kel-F liner during pump operation. Two special inlets with undersized pilot diameters were utilized for this testing. The undersized diameter allowed the inlets to be positioned on the LOX pump volute so that the Kel-F liner was in contact with the impeller.

The procedure for installing the inlet was to push the inlet hard toward the 4 o'clock position so that the Kel-F liner interfered with the impeller wear rings at the 9 to 10 o'clock position. The inlet was then secured (while being pushed against the impeller) by tightening the retaining bolts. All the pump operating conditions except LOX flow were maintained identical to those specified for endurance testing. The LOX pump flow was reduced to 800 to 900 gpm to provide a pressure distribution to force the impeller towards the 9 to 10 o'clock direction, thus providing additional interference force.

Each interference test was conducted for 10 to 15 seconds; the inlet was then removed and examined for evidence of rubbing. Additional rub tests on a particular inlet were achieved by repositioning the inlet so that an unrubbed area contacted the impeller. Only two inlets were reworked with undersized pilot diameters, so it was necessary to replace liners to run all the tests summarized in Table 8.

TABLE 8

COMPONENTS INTERFERENCE TEST STUDY

Inlet Serial No.	LOX Pump Conditions				Rub Depth, inch
	Discharge Pressure, psig	Suction Pressure, psig	Speed, rpm	Flow, gpm	
6	795	52	10,200	840	0.004
6	1050	55	10,100	963	0.006
6	1020	59	9,900	786	0.003
6-2	1050	45	10,400	1245	0.001
6-2	1000	30	10,200	1245	0.001
6-3	1070	67	...	900	0.002
6-3	1070	46	...	900	0.005
3-1	1060	60	9,999	860	0.004
3-1	1040	44	9,900	860	0.004

Nine interference rubbing tests were conducted on four liners. The heaviest rub mark obtained was 0.006 inch deep, and was visible for about 40 degrees around the wide step of the liner.

In general, the liners displayed evidence of rubbing on both steps. The heaviest rubbing occurred on the first step beyond the inducer tunnel section. On lightly rubbed inlets (i.e., rub marks 0.001 inch or less in depth) no evidence of rub marks appeared on the second (largest diameter) step. None of the rubbing tests, including engine tests, revealed any evidence of inducer or of axial rubbing. The majority of the rubbed inlets exhibited rubs from 0.004 to 0.001 inch deep.

Limits Tests

This testing was performed to demonstrate mechanical integrity of the Kel-F-lined inlet when the turbopump performs at its maximum operating conditions. The LOX pump target test conditions were as follows:

Pump Flow, gpm	1350
Pump Discharge Pressure, psig	1100
Pump Suction Pressure, psig	100

These limits were established on the basis of a 10% increase in engine thrust and 18% increase in engine mixture ratio. The pump must operate at a speed of about 11,200 rpm to achieve the LOX pump conditions. At this speed, and utilizing water (which safety required) to replace RP-1 in the fuel pump, the gear box is overpowered and the fuel pump volute is overpressurized. To overcome these difficulties, the fuel pump impeller was reduced in diameter by 1 inch on turbopump S/N R015R.

Nine limits tests were conducted on four inlet assemblies for an accumulated running time of 2700 seconds (Table 9). Four of the tests were conducted before the fuel pump impeller diameter was reduced, and the desired maximum LOX pump conditions were not attained during these tests. Some problems were encountered in adjusting LOX flow and maintaining high suction pressure to the LOX pump, but no other difficulties were encountered with the turbopump or Kel-F-lined LOX inlet.

Internal Pressure Tests

The internal pressure testing was performed on inlet S/N 3, which incorporated specially located pressure taps to monitor pressure within the internal passages of the inlet housing. The purpose of the testing was to determine the magnitude of compressive pressure loading on the liner and to verify that the internal passages were of sufficient size to maintain the pressure at an acceptable level. Three pressure taps were installed in the housing at the following locations:

1. Near the end of the passage that empties back into the pump inlet
2. Between the housing and Kel-F liner near the center of the inducer tunnel section
3. The annular passage adjacent to the bolted flange of the liner

Both normal acceptance and extreme operating conditions were imposed on the turbopump to determine variations of the pressures within the inlet passages. Internal pressures were monitored during five tests. Test 1 (Table 10) represented normal test conditions, and the remaining four tests represented severe operating conditions with respect to increased

TABLE 9

LIMITS TEST SUMMARY

Inlet Serial No.	LOX Pump Conditions				Remarks
	Discharge Pressure, psig	Suction Pressure, psig	Speed, rpm	Flow, gpm	
1	810	..	9,900	1340	
1	840	50	10,203	1340	
4	1080	42	10,800	1240	
4	1070	92	10,500	1240	
8X	1157	95	11,200	1290	Fuel impeller diameter reduced
8X	1110	100	11,300	1390	
8X	1090	95	11,190	1400	
7	1100	100	11,200	1410	
7	1100	95	11,300	1400	

TABLE 10

INTERNAL PRESSURE TEST SUMMARY

Test No.	LOX Pump Conditions				LOX Inlet Pressure, psig		
	Discharge Pressure, psia	Suction Pressure, psia	Speed, rpm	Flow, gpm	Tap Location*		
					1	2	3
1	930	43	9,999	1179	46	46	46
2	975	80	11,199	1440	81	82	82
3	1100	50	10,200	930	..	..	60
4	1060	51	9,999	860	..	..	61
5	1040	44	9,900	860	..	..	58

*For location of inlet pressure taps see Appendix B pressure, and/or flow. In addition to the normal pump instrumentation, all three inlet pressures were monitored and recorded during tests 1 and 2. During tests 3 through 5, the pressure within the annular passage (location 3, Fig. B-3, Appendix B) was monitored and recorded by an oscillograph to establish the existence of pressure variations which were not picked up by the slower response instruments previously used.

Tests 1 and 2 indicated that all three inlet pressures were the same. The inlet pressures reflected the observed variation in suction pressure, but did not reflect variations in pump discharge pressure.

The conclusion drawn from the test data is that there is little or no pressure differential within the inlet that might tend to collapse the Kel-F liner against the inducer or impeller.

An external collapsing force does exist within the bolted flanged region, and is caused by the differential pressure between the pump discharge and reduced pressure through the labyrinth seal. This area is amply supported by the clamping action of the retaining bolts and a steel support ring, so no problems are anticipated with the current design.

Because the pressure increase through the inducer is approximately 50 psi, it is expected that a collapsing pressure could be present only in the area of the inducer inlet where the liner is supported by the inlet housing. Using conservative assumptions, the pressure required to collapse the Kel-F tunnel section, when cold, is approximately 500 psi. Because of this, the pressure levels measured within the liner are considered insignificant and will in no way impair the operation of the

Kel-F-lined inlet. Additionally, the pressure oscillations encountered are too low in amplitude and too short in duration to creep or damage the Kel-F liner.

PERFORMANCE TESTS

All component performance testing, with the exception of several cavitation tests, was conducted at the pump calibration facility. The tests were conducted to establish possible performance variations that might occur as a result of utilizing the Kel-F-lined LOX pump inlet. The performance parameters evaluated were (1) cavitation, (2) discharge pressure, and (3) efficiency.

The tests are outlined in the following paragraphs.

Cavitation

LOX Pump Cavitation. A total of 132 cavitation tests were performed on ten different LOX pump assemblies. Each assembly utilized the standard all-metal inlet and the Kel-F-lined inlet to determine a direct comparison between the two. Identical tests, consisting of two cavitation surveys, were conducted on each inlet. Each survey was made at three flows; nominal (1175 gpm), nominal +10%, and nominal -10%. The LOX pump assembly included the LOX volute, impeller, inducer, and inlet assembly. These assemblies were set up on the pump calibration stand in accordance with the clearances specified on Mark 4 assembly drawings.

At the nominal flow of 1175 gpm and 10,000 rpm, the average critical NPSH of the LOX pump tested with the standard all-metal inlet was 12.4 feet. The average critical NPSH when tested with Kel-F-lined inlet was 12.6 feet, indicating that the cavitation characteristics of the LOX pump remained the same with both inlets.

The ten LOX pumps with all-metal inlets were also tested to determine the water calibration point, and results showed that an average critical NPSH of 18.7 feet was obtained. This represents a correction of 6.3 feet from water to LOX at the nominal flow of 1175 gpm.

A typical cavitation performance survey is shown in Fig. 5. The critical NPSH value is defined by a 2% decrease in discharge pressure. Figure 5 shows this value at 1186 gpm to be approximately 12.2 feet for the pump in this test.

Turbopump Cavitation. The LOX pump cavitation testing performed with the turbopump assembly was conducted to evaluate mechanical integrity of the Kel-F liner when subjected to cavitation during an actual turbopump run. To induce cavitation, the testing involved reducing the LOX pump suction pressure during seven tests. Nominal LOX pump conditions, with the exception of the reduced inlet pressure, were maintained as previously outlined in the section entitled Endurance Testing.

Three tests revealed no cavitation with suction pressures reduced to 5 psig. Three tests with suction pressure from 0 to 3 psig were cut because of the rapid drop in discharge pressure when cavitation began. The final test was performed with a 3-inch-diameter orifice in the suction line. Cavitation proceeded more slowly during this test and

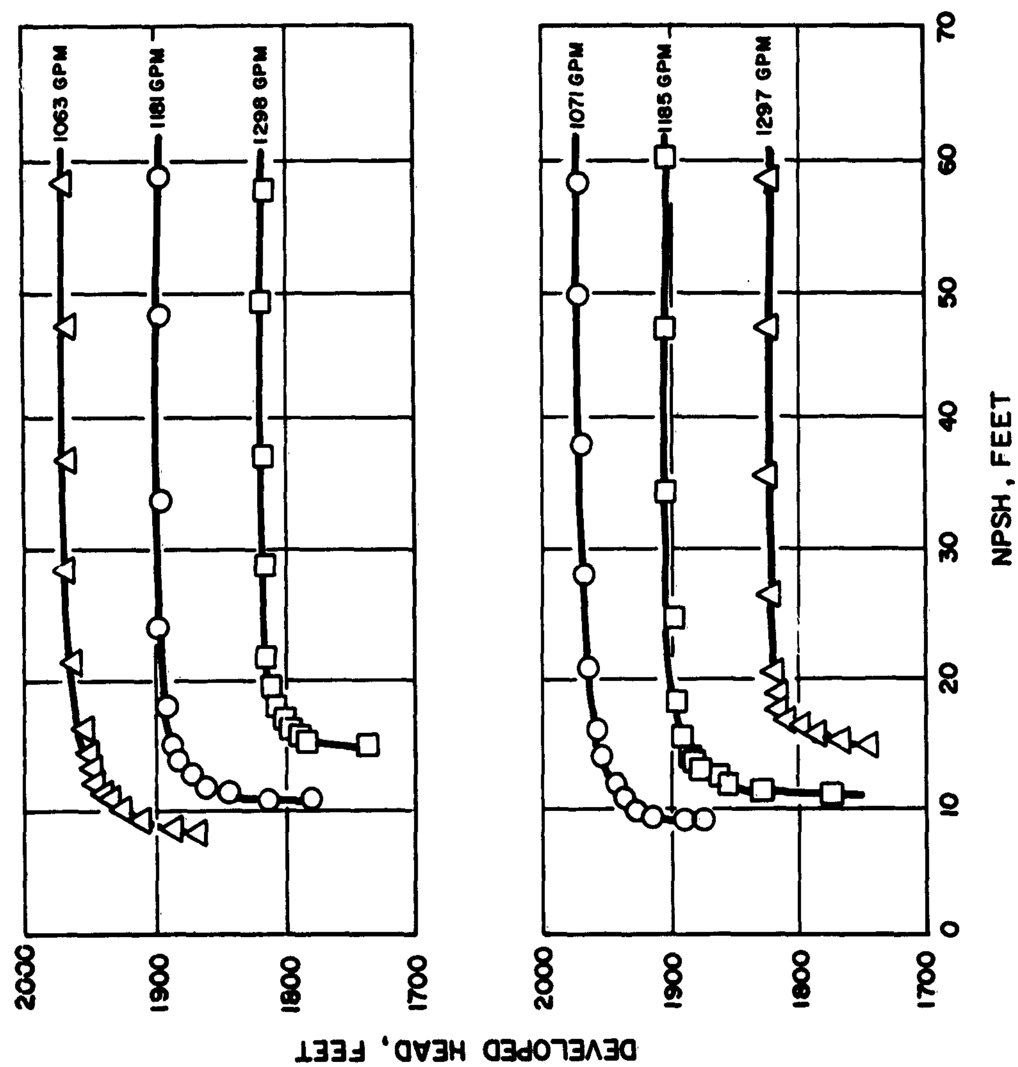

Figure 5. Typical Cavitation Performance Surveys

the turbopump was operated for 2.5 seconds in cavitation. A 10% reduction in discharge pressure was produced. Examination of the Kel-F-lined inlet after these tests revealed no visible damage to the inlet or the pump.

Discharge Pressure

Head vs Capacity. The cavitation performance survey was used to establish head vs capacity performance in the region of high suction pressure (or high NPSH) at which no cavitation is present. This region (Fig. 5) is between 50 and 60 feet of NPSH. Three flows and discharge pressures were obtained to establish the head capacity performance during each survey. The test data show that at the nominal flow of 1175 gpm and 10,000 rpm pump speed the average head rise of the ten pumps tested with the standard all-metal inlet was 1875 feet. The average head rise with the Kel-F-lined inlets was 1901 feet, indicating an increase in head rise for the Kel-F liner of 26 feet.

When determining the water calibration point, the head rise obtained with the standard all-metal inlet in water was 1875 feet, identical to that obtained in LOX. To prevent contamination of the Kel-F-lined assembly, the standard all-metal inlet will be used for water calibrations. The calibration point in water must therefore be reduced by 26 feet to achieve the desired calibration point when using the Kel-F-lined inlet in LOX.

Efficiency

The efficiency data were obtained from two special tests with Kel-F-lined inlets. These two tests were run with inlets S/N 11 and S/N 12 utilizing the same LOX pump components. The hydrodynamic report revealed no change in efficiencies.

Inlet S/N 12. The Kel-F-lined LOX inlet S/N 12 was tested to determine if a rubbed inlet would show any perceptible change in performance. The liner was originally used in S/N 6 inlet, and is the liner which experienced the maximum depth of rub during the interference testing. Six LOX cavitation tests were conducted. Pump speed was 10,000 rpm and flowrates were nominal (1175 gpm), nominal +10%, and nominal -10%. Head vs capacity performance and efficiencies were established by the high suction pressure performance indicated by the cavitation data. The test results revealed no degredation in performance caused by the rubbed area on the liner. The efficiencies measured during this test were between 62 and 64% over the flow range of 1070 to 1290 gpm.

Inlet S/N 11. The Kel-F-lined LOX inlet S/N 11 was tested to determine the performance variation caused by maximum and minimum axial clearances between the impeller and Kel-F liner. Special shims were used to achieve the clearances; the shims were identical in diameter to the inlet-to-housing gasket.

The maximum and minimum axial clearances allowed by the parts tolerances are 0.106 and 0.070 inch, and the actual test clearances were 0.066 and 0.105 inch. The clearance is measured axially between the second impeller wear ring and the corresponding Kel-F liner shoulder (Fig. B-3, Appendix B). Six cavitation tests were conducted at each test clearance. The six tests consisted of two tests at nominal flow, two tests at nominal +10%, and two tests at nominal -10%. Head vs capacity and efficiency were measured at the high suction pressures where cavitation is not present.

The test results showed no change in performance for maximum clearance of 0.105 inch to minimum clearance of 0.066 inch. The efficiency measurements were approximately 61 to 63% over the flow range.

Conclusions

The performance test results revealed a small (2%) increase in head rise for the Kel-F-lined inlet over that of the all-metal inlet. Almost no NPSH change was experienced. Because of the increase in head, future inlets will be calibrated to correct for the increase. The all-metal inlet will be used during water calibrations to preclude the possibility of contaminating the Kel-F-lined inlet assembly. Substitution of the metal inlet during pump calibration is permissible because the proper correction was determined during comparison tests.

Because of the insignificant performance variations between the two inlets, field retrofitting of any Mark 4 turbopump with the Kel-F-lined inlet can be accomplished without recalibrating the LOX pump.

SHAFT DEFLECTION TESTS

SHAFT DEFLECTION TESTS

The objectives of shaft deflection testing were to (1) develop an instrumentation system for measuring shaft deflections, (2) investigate the characteristics of transient and steady-state deflections, (3) investigate characteristics of vibration deflections, and (4) investigate the pressure distribution in the volute of the pump.

The shaft deflection test program was divided into engine-level testing and component-level testing. On the engine level, the following areas were investigated: (1) standard engine sequencing, (2) mixture ratio excursions, (3) valve operating rate variations, (4) shaft deflection frequency, (5) turbopump clearance reductions caused by shaft deflection and (6) effects of mixture ratio on deflection phase angle.

Component testing was divided into the following areas of investigation: (1) oxidizer pump operating point variations, (2) fuel pump operating point variations, (3) oxidizer pump cavitation, (4) air and hydrodynamic pressure tests, and (5) alternate fluid (liquid nitrogen) operation. A comparison was also made between shaft deflection and solder cone tests previously performed with the Atlas MA-3 sustainer turbopump. This comparison is detailed in Appendix D.

MEASURING DEVICE

The device used to measure LOX pump shaft deflections was developed by Rocketdyne and installed between the LOX pump inlet adapter and the low-pressure LOX ducting (Fig. 6 through 8). The fundamental goal of

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Figure 6. Shaft Deflection Instrument

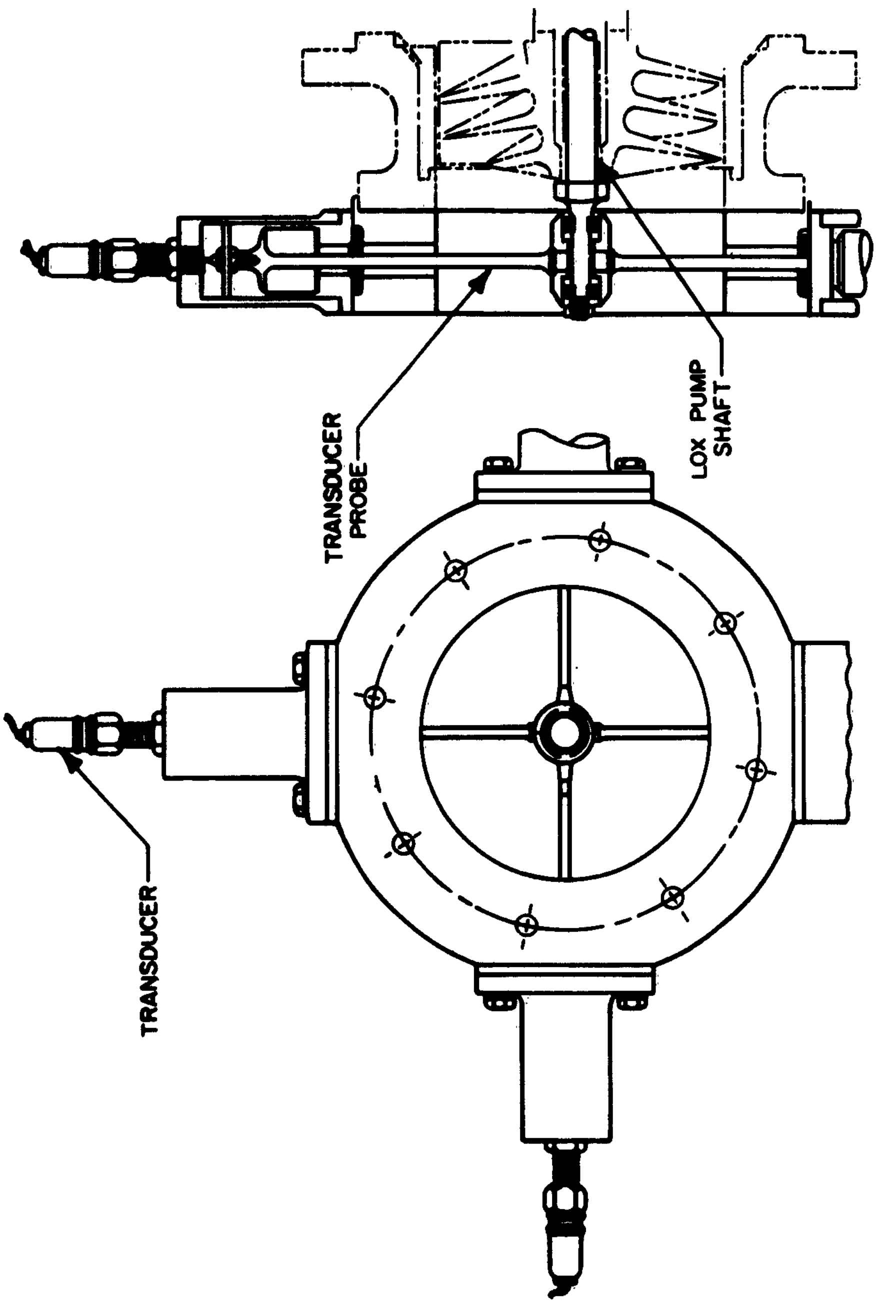

Figure 7. Mechanical Shaft Deflection Instrument Installed on LOX Pump

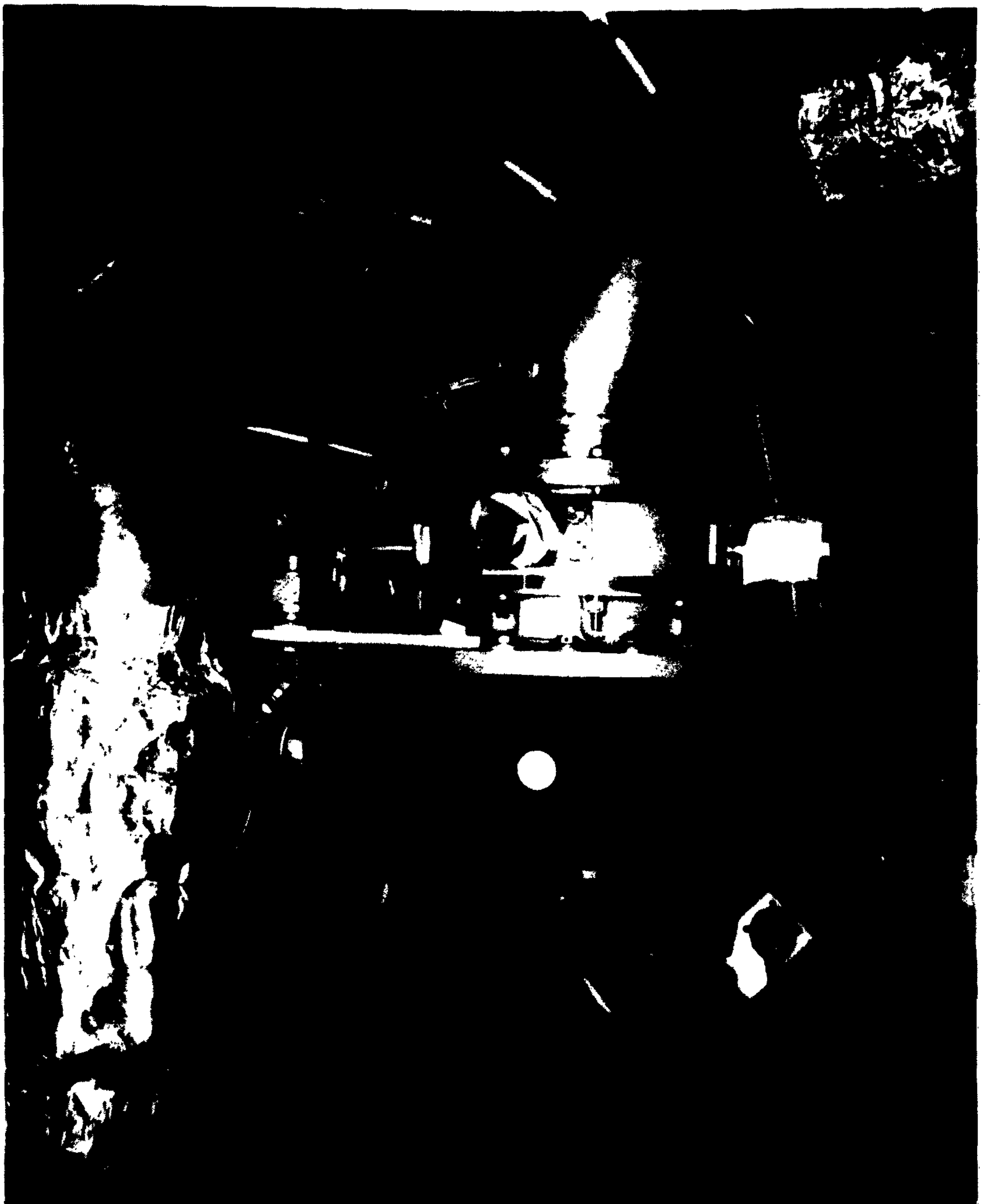

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Figure 8. Shaft Deflection Device Installed on Engine

the program was to measure the deflection of the LOX pump shaft relative to its housing, and the following criteria were established for design of the instrument:

1. All materials must be chemically compatible with LOX.
2. The instrument must not leak and must be structurally sound to the extent that the possibility of a failure which could result in a LOX pump explosion is minimized.
3. Friction must be minimized.
4. A design should be used that employs transducers which give a positive indication of deflection and which have a linear response.
5. The instrument must be insensitive to axial shaft displacement.
6. The instrument must be capable of determining the direction of deflection as well as the magnitude.
7. The instrument must be capable of measuring radial deflections over a range of 0 to 0.050 inch in any direction. (This range was chosen because it exceeds the deflection required to cause rubbing.)
8. The measurements must be independent of shaft speed.
9. The entire instrument system must be capable of recording frequencies over a wide range. (An initial requirement of 0 to 600 cps appeared sufficient, as this was more than triple the shaft rotational frequency range of 170 cps.)
10. The instrument must be capable of withstanding normal turbopump vibrations.

11. The instrument system must demonstrate a high degree of thermal stability at cryogenic temperatures.

12. It must be possible to calibrate the instrumentation system at cryogenic temperatures or to demonstrate a simple correlation between the instrument response at cryogenic conditions and at ambient conditions.

13. The instrument must be compatible with the engine and must allow a variation of engine sequence and off-design turbopump operation.

The system chosen employed linear transducers (Fig. 9) directly actuated by the shaft to give a positive and continuous indication of shaft movement relative to the housing. The orientation of the transducers on the turbopump LOX inlet is shown in Fig. 7 . The retaining bolt for the turbopump inducer was replaced by a special bolt with a shaft extension that served as a spindle for two close-tolerance ball bearings. The bearings operated totally immersed in liquid oxygen and were designed for such service.

The hollow rods that actuated the transducers were pinned to the bearing carrier. The threaded transducer probes were attached to the ends of the hollow rods. The armature portion of the probe was fabricated from magnetic stainless steel and the probe extension was fabricated from nonmagnetic stainless steel.

The housing of the instrument was a ring that was bolted between the pump inlet and the low-pressure LOX duct. The diameter of the inlet flange of the turbopump, which normally has a large tolerance, was machined to give a tight fit for this application.

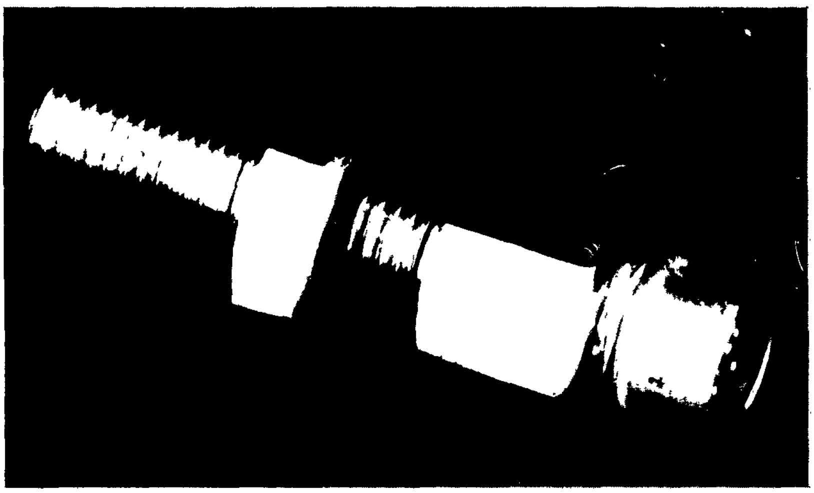


Figure 9 . Shaft Deflection Position Transducer

Each hollow rod passed through a hole in the housing and formed an integral part of the bellows assembly which served as a seal. The mating surfaces of the housing and bellows assembly were sealed with a Naflex seal. Four transducers were equally spaced around the housing ring and mounted on metal caps which bolted to the housing and enclosed the bellows assemblies. A Teflon washer was attached to the bellows assembly to guide the probe and to prevent damage to the probe or bellows from possible side motion.

Each transducer was sensitive only to motion in the direction of its longitudinal axis so that the orthogonal pairs of transducers measured the rectangular components of the resultant shaft deflection. Shaft movements were sensed by a center-tapped variable-reluctance type of position transducer. This transducer transmitted an amplitude-modulated signal upon which the amount of modulation at any given time was proportional to the position of the LOX pump shaft relative to the zero position. The demodulated signal output from the position transducers was recorded on magnetic tape and oscillograph recording instruments.

The Rocketdyne shaft deflection measuring device demonstrated a capability of accurately and continuously measuring position variation in a frequency range of 0 to 600 cps with a maximum amplitude of ±0.050 inch from the null position. The device was limited in this application by the particular transducer range used, but transducers are available to increase the maximum amplitude capability of this device during future similar testing.

The advantage of this measuring device was that it provided a positive mechanical connection between the LOX pump shaft and the outside environment. This connection gave a positive indication of the position of the

LOX pump shaft, was equal in magnitude to the shaft deflection, and was not susceptible to response lag. The system is capable of operation in a liquid oxygen atmosphere under either normal or malfunction conditions and has a minimum effect upon turbopump operation.

Figure 7 shows the installation of this device on the LOX pump and shows the mechanical connection of the position-indicator transducer probe to an extension of the LOX pump shaft. For the engine test series, a 20,000-cps carrier signal was impressed upon the transducer and the instrumentation system was calibrated such that the demodulated signal output from the transducer showed a null when the LOX pump shaft was in its static position. The shaft was exposed to LOX and cooled to its normal operating temperature of approximately -294 F, and the null position of the demodulated transducer output was again found. Each probe was then moved ±0.050 inch from its null position and the output of each transducer at the ± span position was recorded. The demodulated output of the transducer is linear within its recommended usage range, so calibration charts were used to find the position of each probe at any intermediate position between 0 and ±0.050 inch.

The demodulated transducer signal was passed through a 3500-cps bandpass filter to reduce the signal-to-noise ratio of the instrumentation. Data were recorded by magnetic tape and by oscillograph because the oscillograph was response-limited to approximately 350 cps due to the response limitation of its galvanometer. The magnetic tape recorder provided a response far above the 3500-cps limit imposed upon the system by its bandpass filter. A block diagram of the instrumentation system is shown in Fig. 10. Figure 11 presents a typical oscillograph recording taken during a component shaft deflection test.

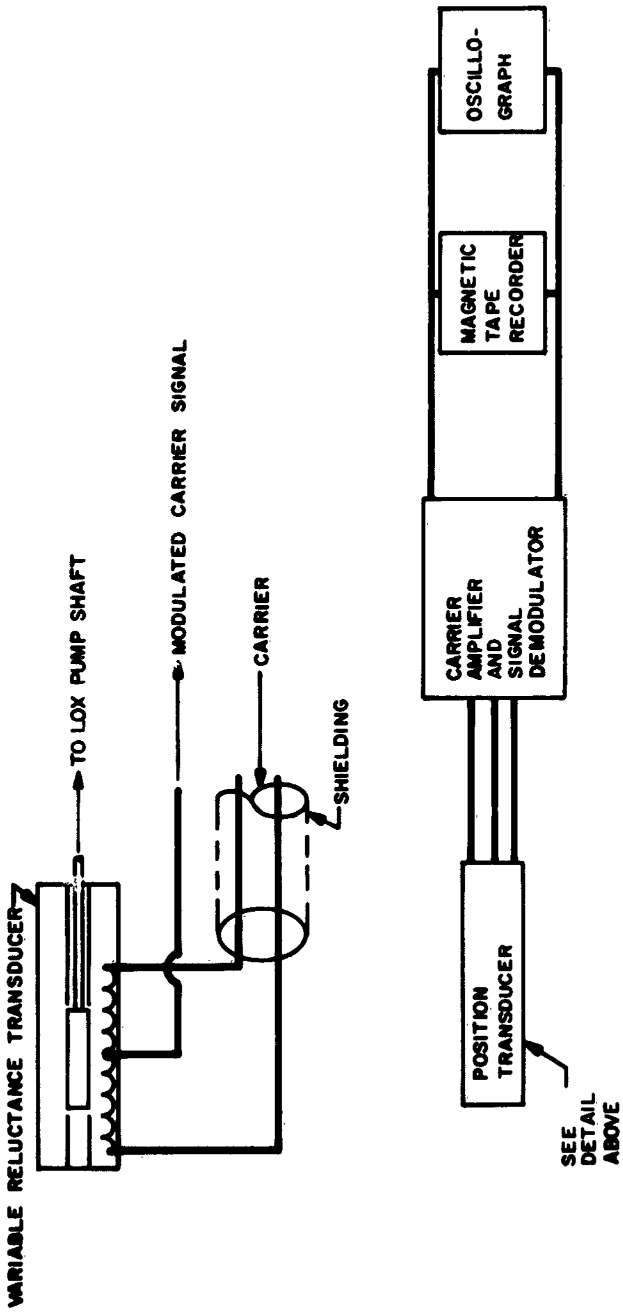


Figure 10. Instrumentation Diagram for LOX Pump Shaft Deflection Instrument

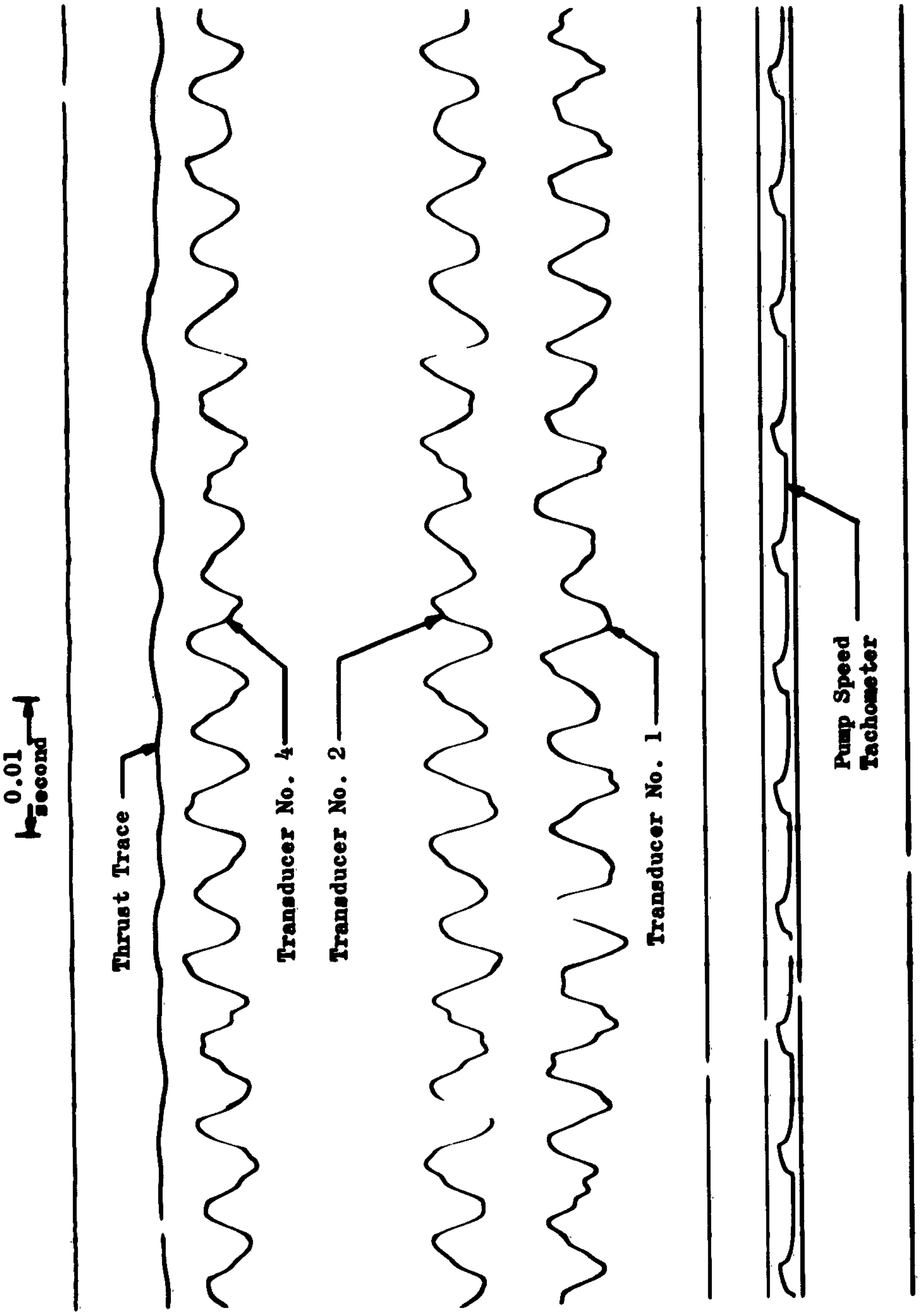


Figure 11. Typical Shaft-Deflection Oscillograph Recording

THEORETICAL CONSIDERATIONS

The theoretical manner by which certain parameters will affect deflections produced in rotating machinery are examined in the following paragraphs.

Figure 12 shows a shaft which is supported by self-aligning bearings at each end. A disk considered to be a concentrated mass weighing W pounds is located at the center of the shaft. If the center of gravity of the concentrated mass is on the axis of rotation, there will be no unbalance to cause the shaft to rotate about an axis other than the axis of the shaft. Unfortunately, such a condition cannot practically be achieved, and the center of gravity of the disk may be assumed to be a small distance (e) from the geometric center of the disk.

Because the center of gravity is away from the axis of rotation and the axis of the bearings, an inertia force causes the shaft to deflect. The deflection at the center of the shaft is represented by (r) in Fig. 12b. The geometric center of the disk (O) is the same as the center of the shaft at the disk.

As the shaft rotates, the high spot (T) will rotate about the axis (S) of the bearings. The inertia force of the disk is balanced by what may be called the spring force of the shaft as the shaft rotates. The inertia force for a mass rotating about a fixed center is $W/g\ (r + e)\ \omega^2$, where ω is the angular velocity and g is the gravitational constant.

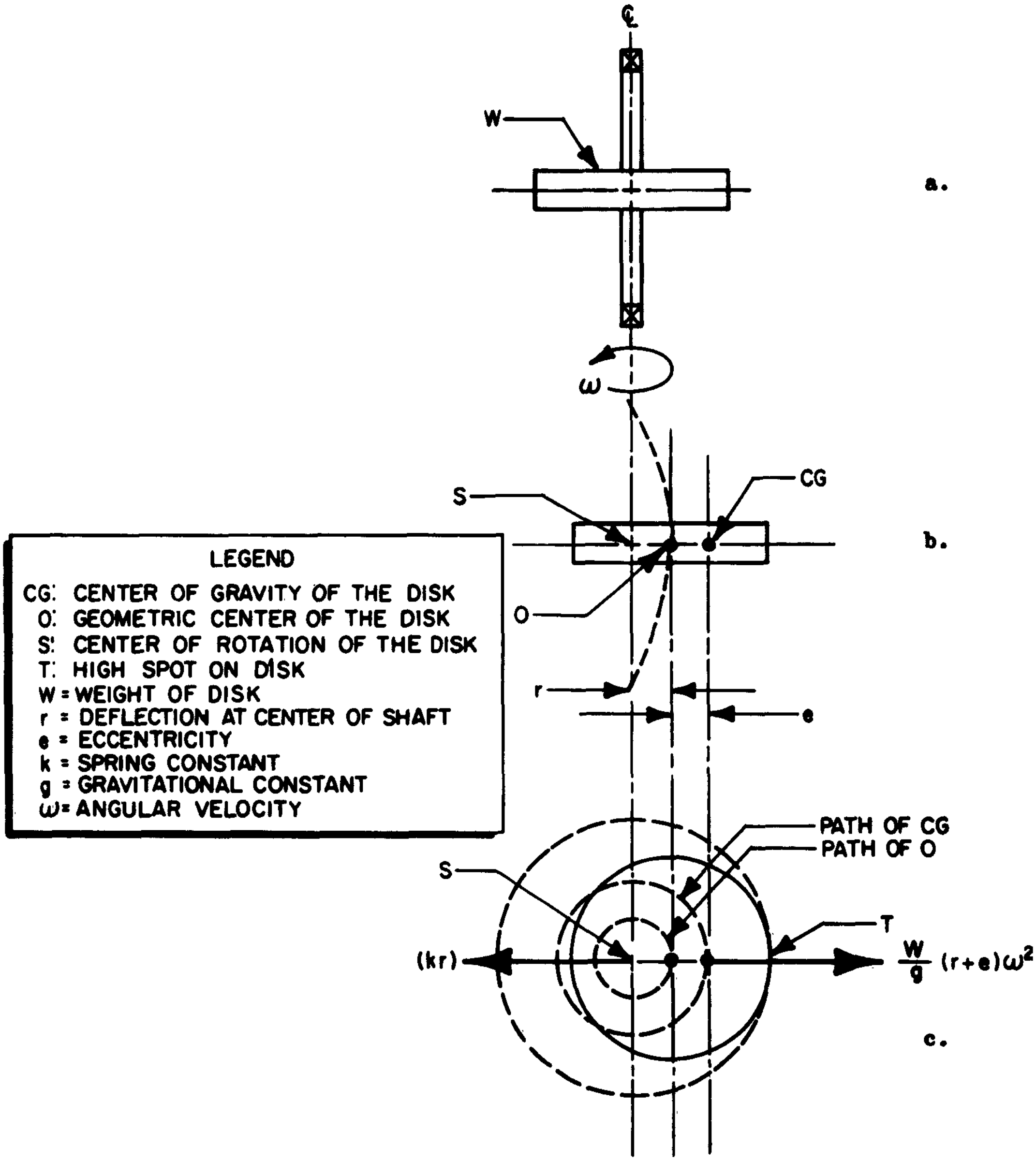


Figure 12. Theoretical Motion and Forces Applied To a Typical Shaft and Disk

The spring force of the shaft may be represented by (kr), where (k) is the shaft spring rate in pounds required for each inch of deflection of the shaft at the disk. Summing up the forces at the disk in Fig. 12c equal to zero, we obtain:

$$\frac{W}{g}(r + e)\,\omega^2 - kr = 0$$

Rearranging terms,

$$\frac{r}{e} = \frac{\frac{W}{g}\,\omega^2}{k - \frac{W}{g}\,\omega^2}$$

Figure 13 shows a typical plot of r/e vs ω, with the effects of friction included, and reveals that when a constant coefficient of friction is presumed, the ratio of r/e becomes larger as ω/ω_n (where ω_n is the critical speed) approaches a value of 1.0.

From this simplified analysis of dynamic shaft deflection, the following factors can be expected to influence the magnitude of the dynamic deflections of the turbopump shaft, impellers, and gear assembly:

1. The unbalance of the assembly
2. The mass of the assembly
3. The critical speed of the assembly
4. The elastic properties of the assembly
5. The operating speed
6. The amount of viscous damping or friction

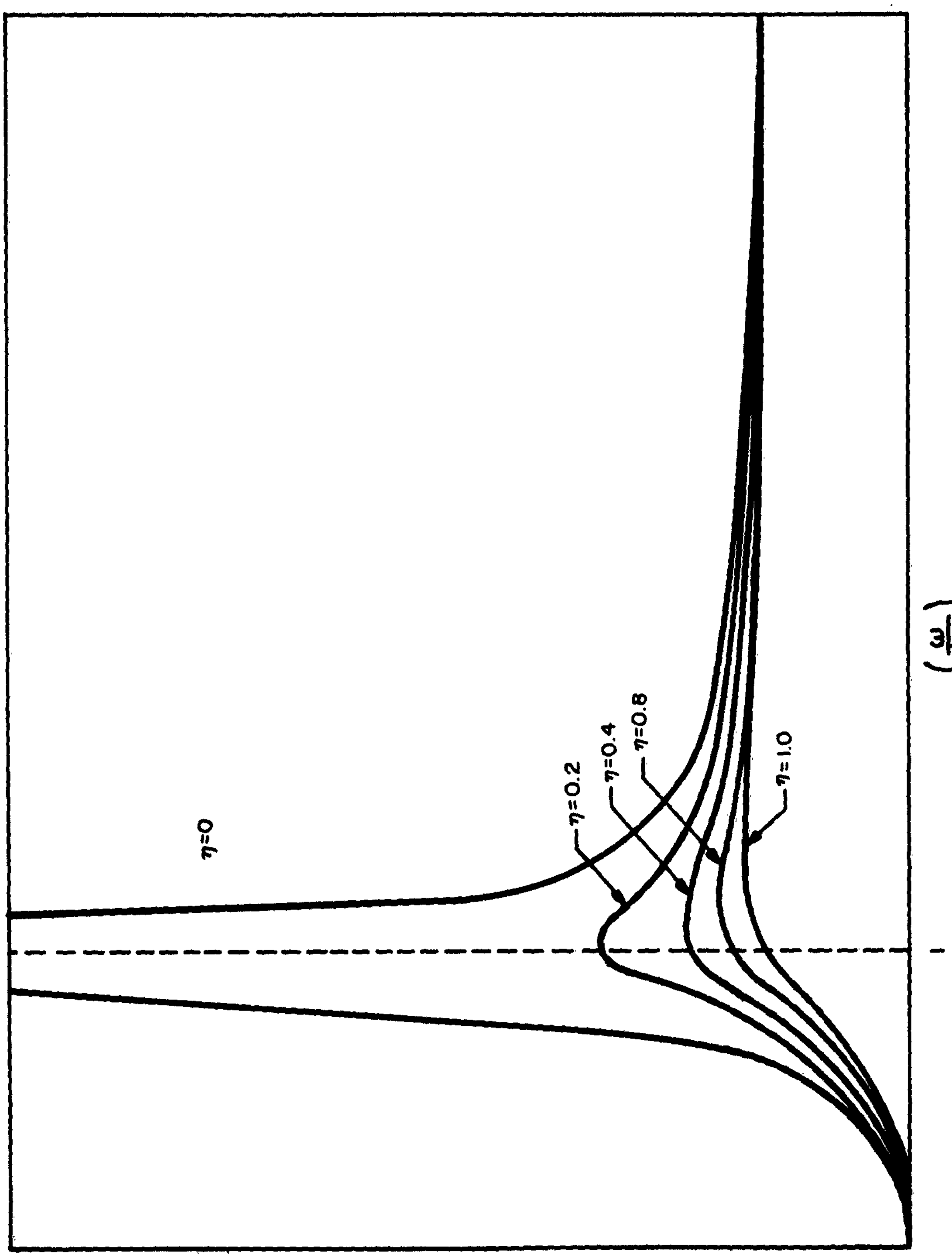


Figure 13. Effect of Speed and Friction on Vibration Amplitude

A detailed analysis of the critical speed of the Mark 4 rotating assembly, based on results of an analog computer study, is presented in Appendix C. Also included is an analysis and prediction of the magnitudes of the vibration or dynamic deflections which may be experienced by the oxidizer impeller under various conditions of unbalance, operating speed, and ball bearing spring rates. The lowest critical speed determined was 15,200 rpm, whereas the maximum turbopump operating speed is 11,200 rpm, or 75% of the lowest critical speed.

The greatest deflection calculated in the plane of the static loads was approximately 0.0074 inch for a LOX impeller unbalance of 0.5 in.-oz at 11,200 rpm. An unbalance of 0.5 in.-oz corresponds to an offset of the center of gravity of the LOX impeller of 0.003 inch from the principal axis of the mass and is considered a realistic value for the actual rotating assembly unbalance.

In addition to a dynamic or vibration deflection effect, the shaft can exhibit a static deflection caused by an unbalance of the hydrodynamic forces in the volute of the pump. The volute of a centrifugal pump can be designed for hydrodynamic balance at one operating point only, so it would be expected that a pressure gradient (and, therefore, an unbalanced force) would exist at every other operating point. This unbalanced pressure force will cause a pump shaft deflection that will follow the laws for static loading and corresponding deflections in beams.

In conclusion, the deflection of the oxidizer pump shaft can be thought of as being a summation of the two components discussed above; a dynamic or vibration deflection and a static deflection arising from the pressure unbalance in the oxidizer pump and the fuel pump, and the gear load. Test experience has demonstrated that this analysis accurately describes the deflection of the shaft. Figure 14 presents a plot of shaft deflection at the instrument and at the wear ring vs the load applied to the impeller, as determined by laboratory tests. It is by this relationship that deflections at the instrument were converted to deflections at the wear ring.

ENGINE TEST CONDITIONS

Ten MA-5 engine tests were conducted during the shaft deflection test program, and 981 seconds of engine test time were accumulated. Table 11 summarizes the results of these tests.

Standard MA-5 Engine Hardware

Four of the MA-5 engine tests, totaling 160 seconds of engine test time, were conducted with the standard production type of MA-5 hardware. The objectives of these four tests were to check out the engine with the shaft deflection instrument installed, to check out and calibrate the shaft deflection device, and to measure Mark 4 LOX pump shaft deflections under normal operating conditions including mixture ratio excursions.

Fixed Propellant Utilization System

Five MA-5 engine tests, totaling 810 seconds, were conducted with a fixed propellant utilization (PU) system. The system was fixed by replacing the standard PU servocontrol valve with a valve pad sealing plate. This plate routed hydraulic pressure directly to the closing

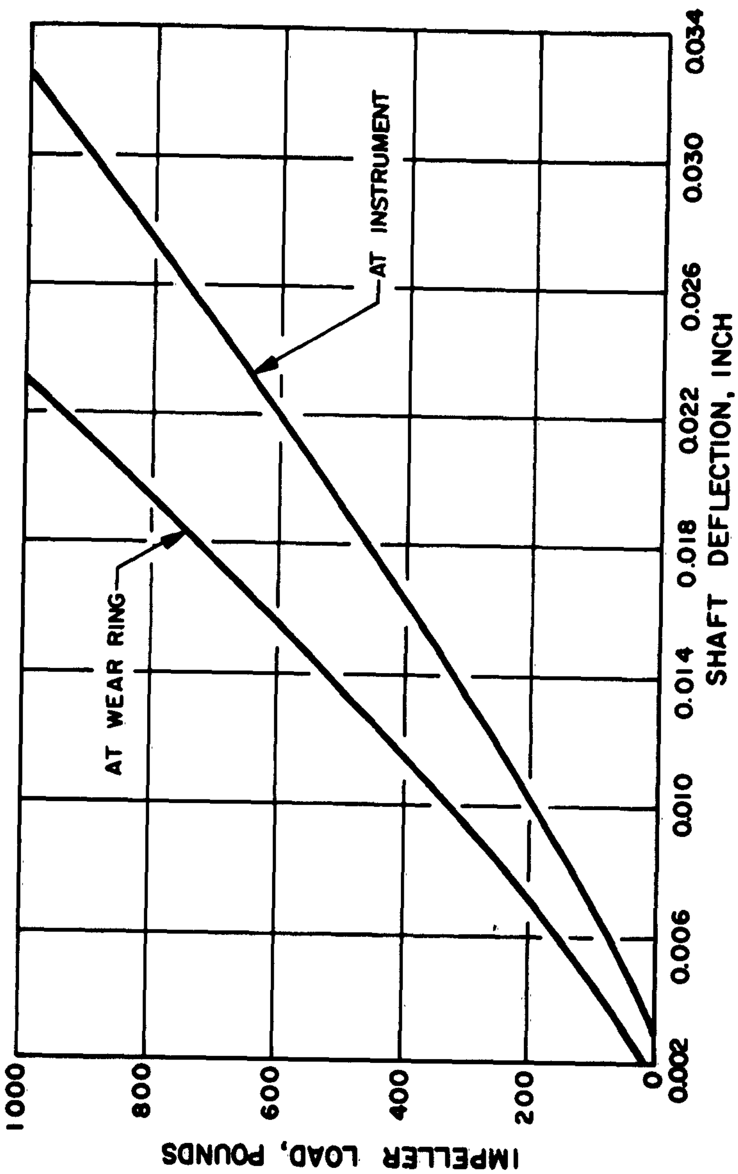


Figure 14. Impeller Load vs Shaft Deflection

TABLE 11

SHAFT DEFLECTION ENGIN

Test No.	Engine Serial No.	Turbopump Serial No.	Test Duration, Seconds	Accumulated Duration, Seconds	Lin Seria
512-137	022-4	4044192	10	10	9X
512-138	022-4	4044192	10	20	9X
513-001	225501	4039905	70	90	9X
513-002	225501	4039905	70	160	9X
513-003	225501	4039905	200	360	9X
513-004	225501	4039905	200	560	9X
513-005	225501	4039905	200	760	9X
513-006	225501	4039905	200	960	9X
513-007	225501	4039905	11	971	6-
513-008	225501	4039905	10	981	6-

*Fixed propellant utilization system

1

TABLE 11

DEFLECTION ENGINE TEST CONFIGURATIONS

mulated ation, conds	Liner Serial No.	Mixture Ratio Controller Part No.	Propellant Utilization Controller Part No.	Propellant Utilization Valve Excursion	Rub
10	9X	250944 (MA-5)	250815 (MA-5)	No	No
20	9X	250944 (MA-5)	250815 (MA-5)	No	No
90	9X	250944 (MA-5)	250815 (MA-5)	Yes	No
160	9X	250944 (MA-5)	250815 (MA-5)	Yes	No
60	9X	250944 (MA-5)	601031 (MA-2)*	No	No
60	9X	250944 (MA-5)	601031 (MA-2)*	No	No
60	9X	250061 (MA-2)	601031 (MA-2)*	No	No
60	9X	250061 (MA-2)	601031 (MA-2)*	No	No
71	6-3	250061 (MA-2)	601031 (MA-2)*	No	Not determined
81	6-3	250944 (MA-5)	250815 (MA-5)	No	Yes

side of the PU valve from the high-pressure manifold of the hydraulic control package. This modification affected the engine as though the PU servovalve had shuttled to its full-closing position. The modification resulted in a considerably reduced PU valve closing time; i.e., the valve traveled from full-open to its position against the hydraulic stop mechanism within 400 milliseconds. The difference between these two valve closing rates is presented in Fig. 24, in a later section of this report. The objective of the tests was to measure LOX pump shaft deflections with an increased PU valve control rate.

Fixed Propellant Utilization System and MA-2 Mixture Ratio Controller. Three of the tests conducted with the fixed PU system also incorporated an MA-2 engine mixture ratio controller in place of the standard MA-5 mixture ratio controller. The objective of these tests was to measure LOX pump shaft deflections under conditions of increased propellant utilization valve and head suppression valve control rates. The MA-2 mixture ratio controller caused the head suppression valve to come into control approximately twice as fast as it would with the standard MA-5 type mixture ratio controller.

Offset Liner to Induce Rubbing

Two tests, totaling 20 seconds of engine test time, were conducted with an offset Kel-F-lined inlet adapter installation in the LOX pump. One of these tests was performed in conjunction with a standard hardware test previously explained. Prior to these tests, the LOX pump inlet adapter was pushed toward the 6 o'clock position (12 o'clock position was the LOX seal cavity drain line) until it interfered with the impeller. The adapter was then tightened down on the LOX pump volute. An analysis of preliminary shaft deflection results indicated that rubbing should be expected between the impeller and the Kel-F liner during these tests.

COMPONENT TEST CONDITIONS

Forty-nine component tests were conducted during the shaft deflection program, and 4735 seconds of test time were accumulated. Below is a summary of the component-level testing.

Purpose of Test	Tests Conducted With LOX		Tests Conducted With Liquid Nitrogen		Total	
	Number of Tests	Total Time Accumulated, seconds	Number of Tests	Total Time Accumulated, seconds	Tests	Time Accumulated, seconds
Shaft Deflection Instrument Development	17	1452	6	782	23	2234
Shaft Deflection Analysis	21	2087	5	414	26	2501
Total	38	3539	11	1196	49	4735

As shown on the previous page, the first 23 tests (Table 12) were directed toward evaluating the design of the shaft deflection instrument, establishing the nature and extent of the recording instrumentation required to support the program, and formalizing the instrumentation operating procudure. The final 26 tests (Table 13) were directed toward an analysis of shaft deflection.

In addition to the above tests, a Mark 4 turbopump was operated with air as the test fluid in the LOX pump and water as the test fluid in the fuel pump. The results were corrected to LOX and RP-1 densities and compared to other component-level tests. During the testing, pump discharge pressures were measured with total-pressure probes located at the centerline of the discharge ducts 40 inches from the discharge flanges. Radial pressures were measured by eight equally spaced static pressure taps located around each pump volute. Flow was measured by flow nozzles capable of attaining 120% of the engine operating flowrate. All pressure and flow measurements were read on water manometers. Pump speed was measured by a magnetic pickup.

INSTRUMENT DEVELOPMENT TEST RESULTS

The initial instrument development test, which did not incorporate the recording instrumentation, demonstrated the mechanical integrity of the design.

The first two tests incorporating the recording instrumentation resulted in a change involving the transducer probes and carrier amplifiers. The original probes were replaced because they did not satisfy the material

TABLE 12

SHAFT DEFLECTION INSTRUMENT DEVELOPMENT TESTS

Test No.	Test Duration, seconds	Turbopump Serial No.	Test Fluid
187	300	R082R	Liquid Nitrogen
218	30	R015R	Liquid Nitrogen
219	300	R015R	LOX
227	225	R005R	Liquid Nitrogen
228	100	R005R	Liquid Nitrogen
229	100	R005R	Liquid Nitrogen
230	300	R005R	LOX
231	100	R005R	LOX
232	100	R005R	LOX
233	39	R005R	LOX
234	31	R005R	LOX
235	45	R005R	LOX
236	43	R005R	LOX
237	43	R005R	LOX
238	31	R005R	LOX
239	27	R005R	Liquid Nitrogen
240	27	R005R	LOX
241	33	R005R	LOX
242	100	R004R	LOX
243	100	R004R	LOX
244	100	R004R	LOX
245	30	R004R	LOX
246	30	R004R	LOX

TABLE 13

SHAFT DEFLECTION ANALYSIS TESTS

Test No.	Test Duration, Seconds	Turbopump Serial No.	Test Fluid
247	70	R004R	LOX
248	33	R004R	LOX
249	41	R004R	LOX
250	25	R004R	LOX
251	20	R004R	LOX
252	75	R004R	LOX
253	110	R004R	LOX
254	70	R004R	LOX
255	56	R004R	Liquid Nitrogen
256	71	R004R	Liquid Nitrogen
257	62	R004R	Liquid Nitrogen
258	65	R004R	LOX
259	65	R004R	LOX
260	65	R004R	LOX
261	180	R004R	LOX
262	60	R004R	LOX
001	109	R004R	LOX
002	120	R004R	LOX
003	140	R004R	LOX
004	120	R004R	LOX
005	75	R004R	Liquid Nitrogen
006	211	R005R	LOX
007	171	R005R	LOX
008	199	R005R	LOX
009	138	R005R	LOX
010	150	R005R	Liquid Nitrogen

and dimensional specifications set forth by the transducer manufacturer. These discrepancies caused a reduced and distorted transducer signal. The original 10,000-cps carrier amplifiers were replaced by 3000-cps carrier amplifiers. Because of considerable previous experience with the 3000-cps carrier amplifiers, some of the unknown variables were eliminated and the problem analysis was simplified. Subsequent engine tests using a 20,000-cps carrier system indicated that the 10,000-cps carrier system would have been satisfactory.

As a result of the development testing, a velocity transducer and a bell-crank arrangement used with a transducer were eliminated. The velocity transducer was discarded when it was learned that the data were redundant and did not assist in expediting the analysis. The bell-crank arrangement was eliminated when the formation of ice caused the ball joints to lock. This arrangement had been designed to facilitate the use of limited space on the engine. Development established the feasibility of using only three similar transducer configurations for engine tests. Four similar transducers were used during shaft deflection analysis component tests because performance could be verified by diametrically opposed transducers.

An apparent zero shift between pretest static and posttest static conditions was a major problem throughout the development testing. Investigations were made to determine if the zero shifts were mechanical or electrical. The transducer signal voltages and temperatures were observed during the chilldown period with inconclusive results. The tests demonstrated that the transducer signal voltages and temperatures were unstable for the first 30 minutes of chilldown; this limit was less than the chilldown time experienced by the turbopump during some of the tests which had demonstrated zero shift. Though no definite cause for the lack of shift was established, an awareness of the critical nature of the childown fostered rigid adherence to the operating procedure and resulted in a reduction of the frequency of occurrence to an acceptable level.

DATA REDUCTION METHOD

Engine test data reduction was accomplished by transferring analog data from oscillographic and magnetic tape recordings onto punched cards which were used in conjunction with a digital computer data reduction program. Data points were taken every millisecond over a 400-millisecond time slice. Data from the individual position transducers were automatically reduced to yield the position of the turbopump shaft at succeeding 1-millisecond intervals. The shaft position for a given time point was then automatically plotted by computer peripheral equipment on polar coordinates and on a time scale.

SHAFT DEFLECTION TEST RESULTS

Standard Engine Sequencing

Figure 15 shows a comparison of various pump parameters and the static component of shaft deflection during the engine start transient period for test 512-137, and Fig. 16 presents a standard MA-5 start sequence. Figure 15 reveals that the static component of the shaft deflection increases during the ignition stage and initial pump acceleration. During test 512-137, the deflection reached a maximum start transient value of 0.016 inch at the instrument (comparable to 0.011 inch at the wear ring) during stabilized ignition-stage operation from the start tanks. The start transient deflection noted during this test was the maximum recorded during any of the ten shaft deflection tests.

As the propellant utilization valve opens, additional fuel flows into the chamber and thrust buildup begins. As thrust buildup progresses, bootstrap occurs, oxidizer flow increases and the pump accelerates a second time. When the oxidizer flow per revolution of the pump shaft (represented by Q_{LOX}/N) decreases, the static shaft deflection also decreases. Static shaft deflection is a minimum start transient value when Q_{LOX}/N reaches its minimum start transient value. The Q_{LOX}/N nominal design operating point is 0.117 gallon/revolution.

During test 512-137, static shaft deflection reached a minimum point when Q_{LOX}/N had a value of 0.117 gallon/revolution. As Q_{LOX}/N increased from its minimum point during thrust buildup, shaft deflection increased to a second peak and leveled off. The characteristic of peaking and then dropping off to mainstage values is seen on Fig. 15, which tends to verify the theoretical prediction that shaft deflection is affected by oxidizer volumetric flowrate and pump speed.

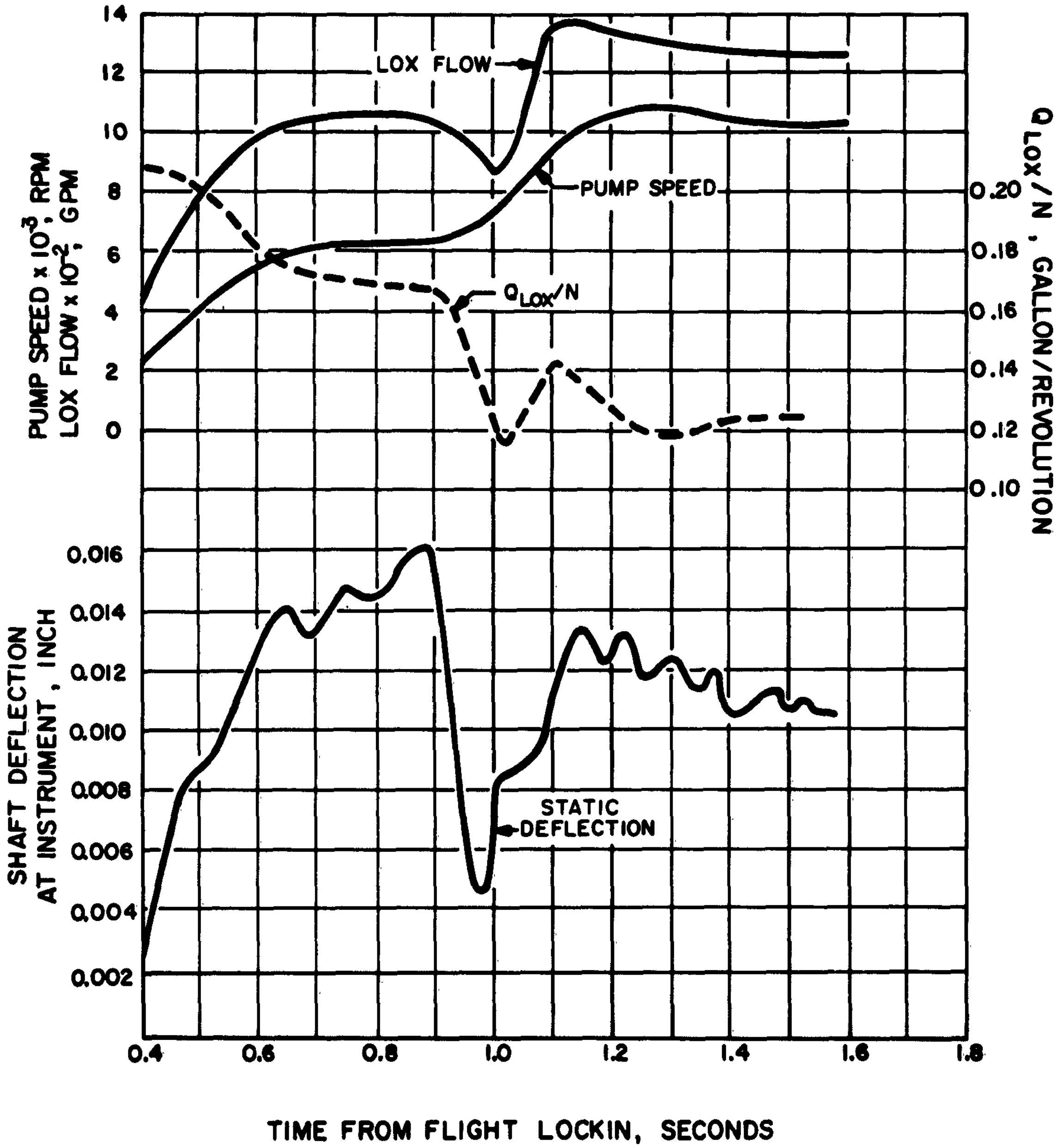


Figure 15. Pump Speed, LOX Flow, Q_{LOX}/N, and Shaft Deflection After Flight Lockin, Test 512-137

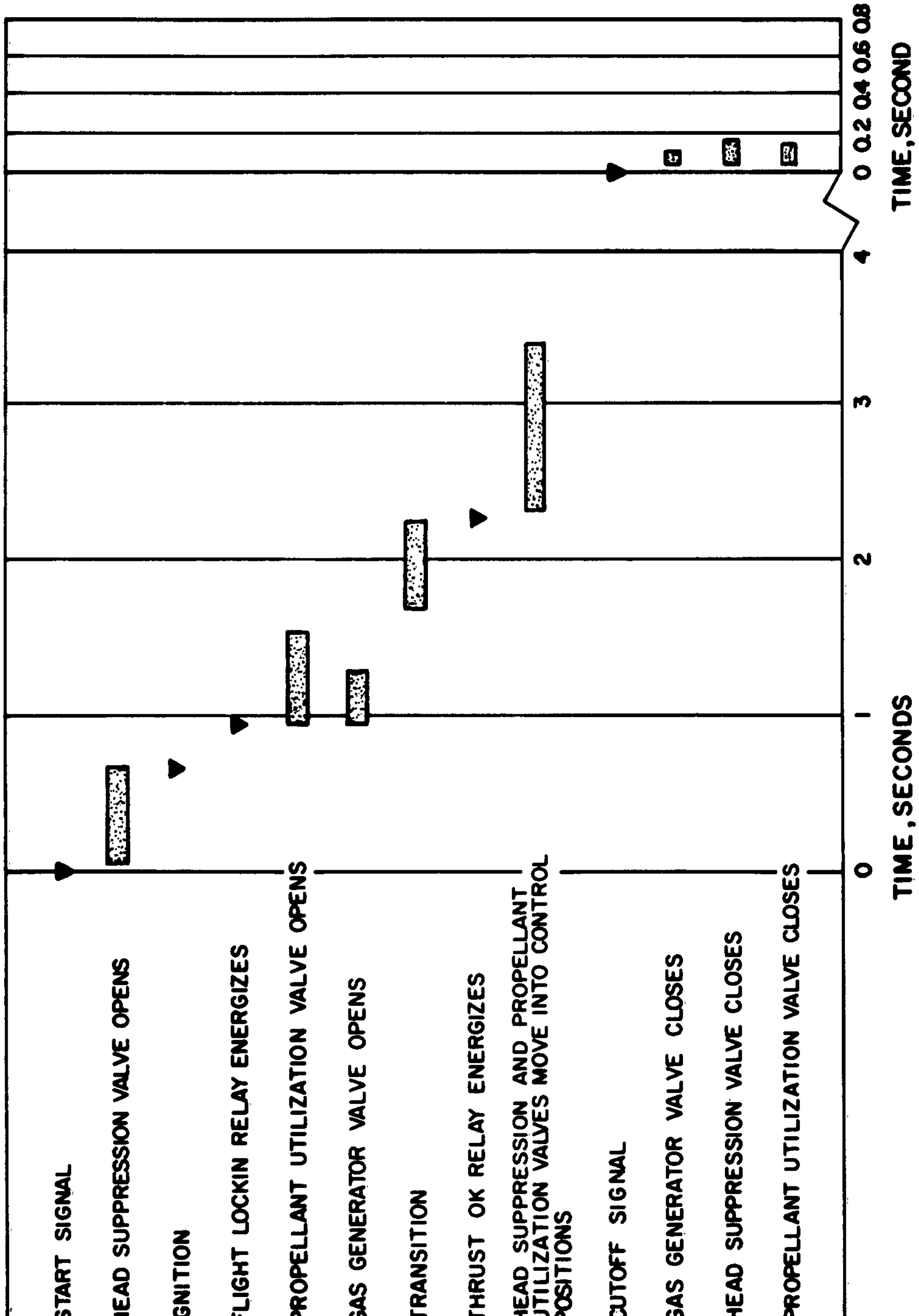


Figure 16. Standard Engine Sequencing

Figure 17 presents values of static shaft deflection during start transient for test 513-004. Tests 512-137 and 512-138 were conducted with turbopump S/N 4044192 on engine S/N 022-4, and tests 513-001 through 513-008 were conducted with turbopump S/N 4039905 on engine S/N 225501. Although the start transient deflections during tests 512-137 and 512-138 agreed in magnitude and in general shape of the curve, as did the start transient deflections during tests 513-001 through 513-008, the start transient deflection characteristics between the two turbopumps were not in close agreement. Each pump experienced a characteristic decrease in shaft deflection; this occurred simultaneously with the dip in Q_{LOX}/N. The start transient static deflection generally was higher on turbopump S/N 4044192 than on pump S/N 4039905, and the greatest static deflection experienced during start transient on pump S/N 4044192 was 0.011 inch at the wear ring. The largest start transient static deflection observed on pump S/N 4039905 was 0.009 inch at the wear ring.

Vibration or dynamic deflection during start transient was lower in amplitude than mainstage values of vibration deflection. The average vibration during transient operation just prior to thrust buildup was 0.004 inch peak-to-peak at the wear ring, while the average peak-to-peak amplitude of vibration during mainstage operation at nominal mixture ratio was 0.007 inch. These affects would be predicted by theory, since the turbopump speed is lower during start transient than it is during mainstage.

The shaft deflection data revealed an increase in static shaft deflection during cutoff. Figure 18 presents static deflection data after cutoff on test 512-137, and shows the maximum deflection to be 0.0205 inch at the instrument. This corresponds to a deflection of 0.0146 inch at the wear ring. Figure 19 shows a polar plot of shaft deflection during engine start and cutoff.

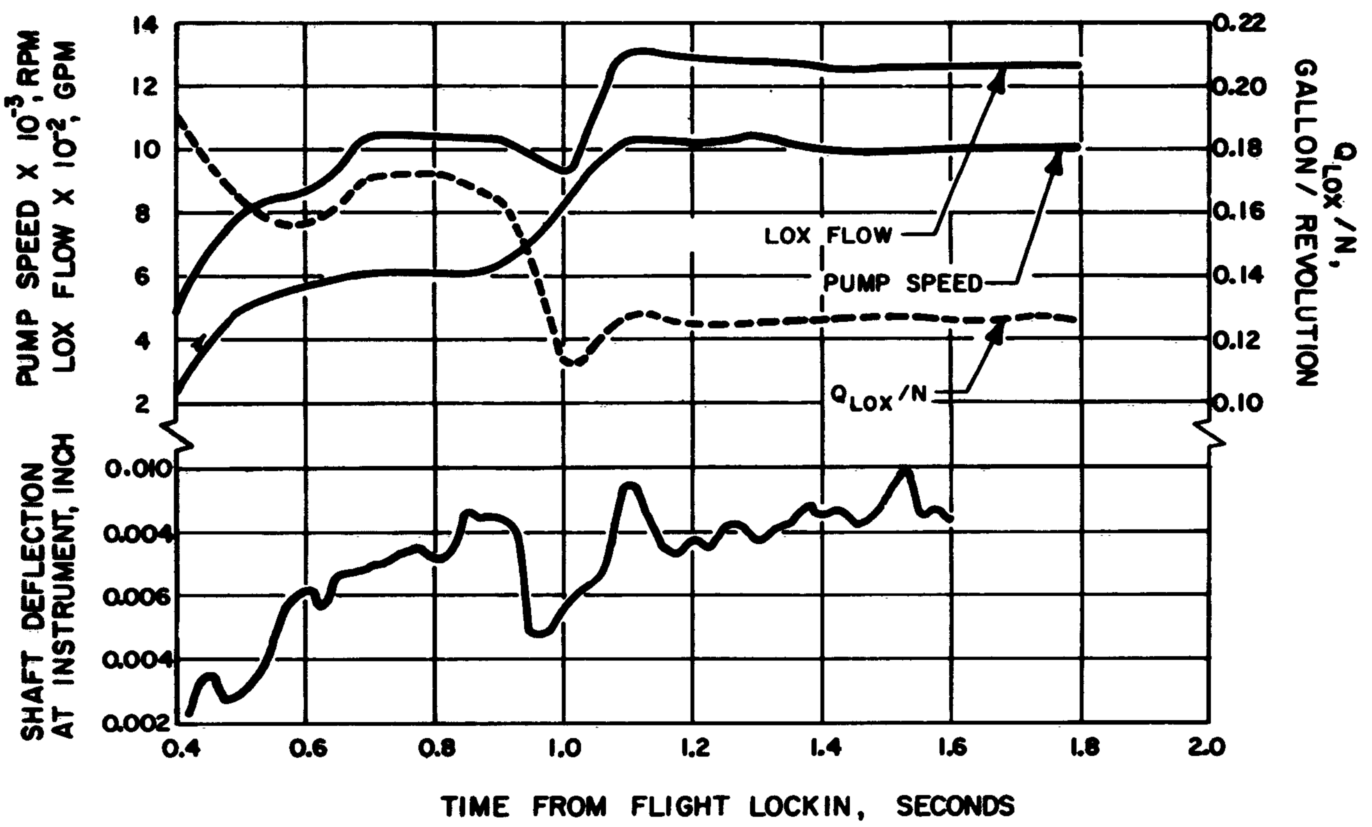


Figure 17. Pump Speed, LOX Flow, Q_{LOX}/N, and Shaft Deflection After Flight Lockin, Test 513-004

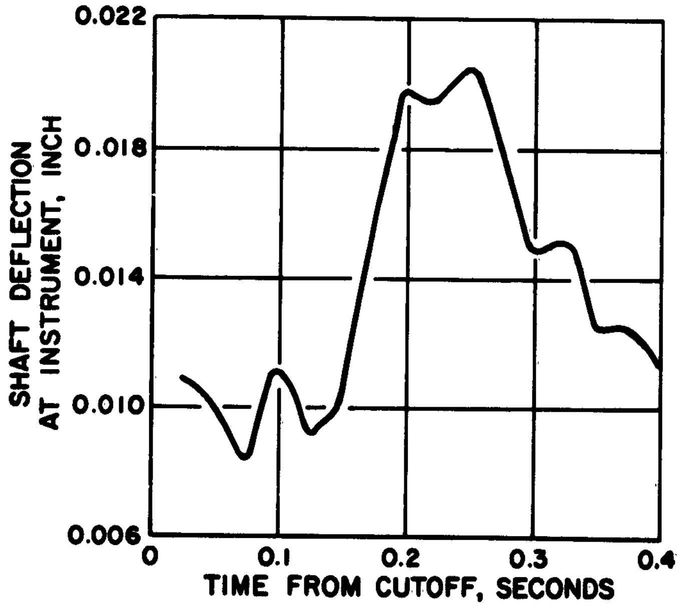
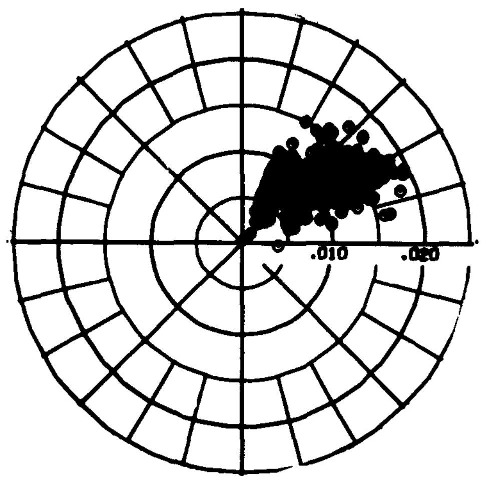


Figure 18. Static Shaft Deflection After Cutoff, Test 512-137

0.4 to 0.8

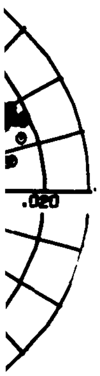
Time from Flight Lockin, seconds



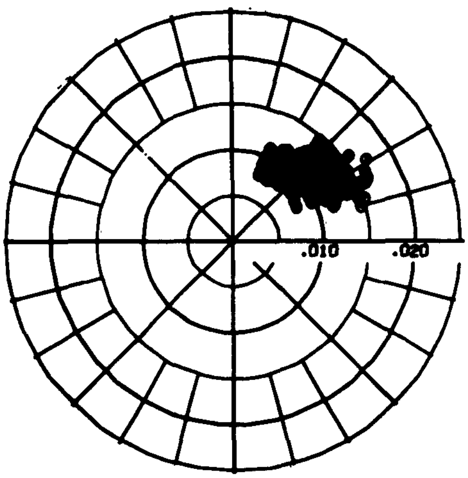
0.8 to 1.2

Time from Flight Lockin, seconds

1

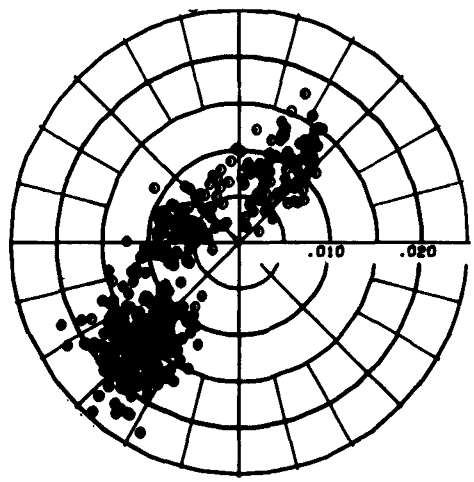


conds



1.2 to 1.6

Time from Flight Lockin, seconds



0 to 0.4

Time from Cutoff, seconds

Figure 19. Shaft Position During Start and Cutoff

Mixture Ratio Excursions

Tests 513-001 and 513-002 were conducted with the engine operated at ±15% of nominal mixture ratio. The objective of these tests was to determine if varying the pump hydrodynamic operating point would affect the amount of deflection. As engine mixture ratio is increased, Q_{LOX}/N increases and N (pump speed) decreases. The reverse occurs as engine mixture ratio is decreased. Figure 20 shows how static deflection is affected by varying the LOX pump operating point. This curve shows that the static deflection for a given pump is a minimum at the pump design point (0.117 gallon/revolution) and increases as (Q_{LOX}/N) varies in either direction from the design point. Curves are drawn for sustainer engine S/N 225501 (turbopump S/N 4039905) and for turbopumps S/N R004R and R005R which were used during the component test series. The displacement of the curves from each other can be attributed to pump-to-pump variations. Increased static deflection is expected to occur when the operating point varies from the design point because, as the operating point changes, a pressure gradient is produced around the periphery of the volute and an associated unbalanced force is applied to the impeller. The pressure gradient and the unbalanced force increase in magnitude as the distance increases between the operating point and the design point, and static deflection increases with an increase in the unbalanced force.

Based on engine S/N 225501 data the static component of the shaft deflection at design point operation can be expected to be approximately 0.0082 inch the wear ring. At ±15% mixture ratio, 0.0102 inch of static deflection was measured at the wear ring and at -15% mixture ratio, 0.0088 inch of static deflection was measured at the wear ring. These data are tabulated on page 94.

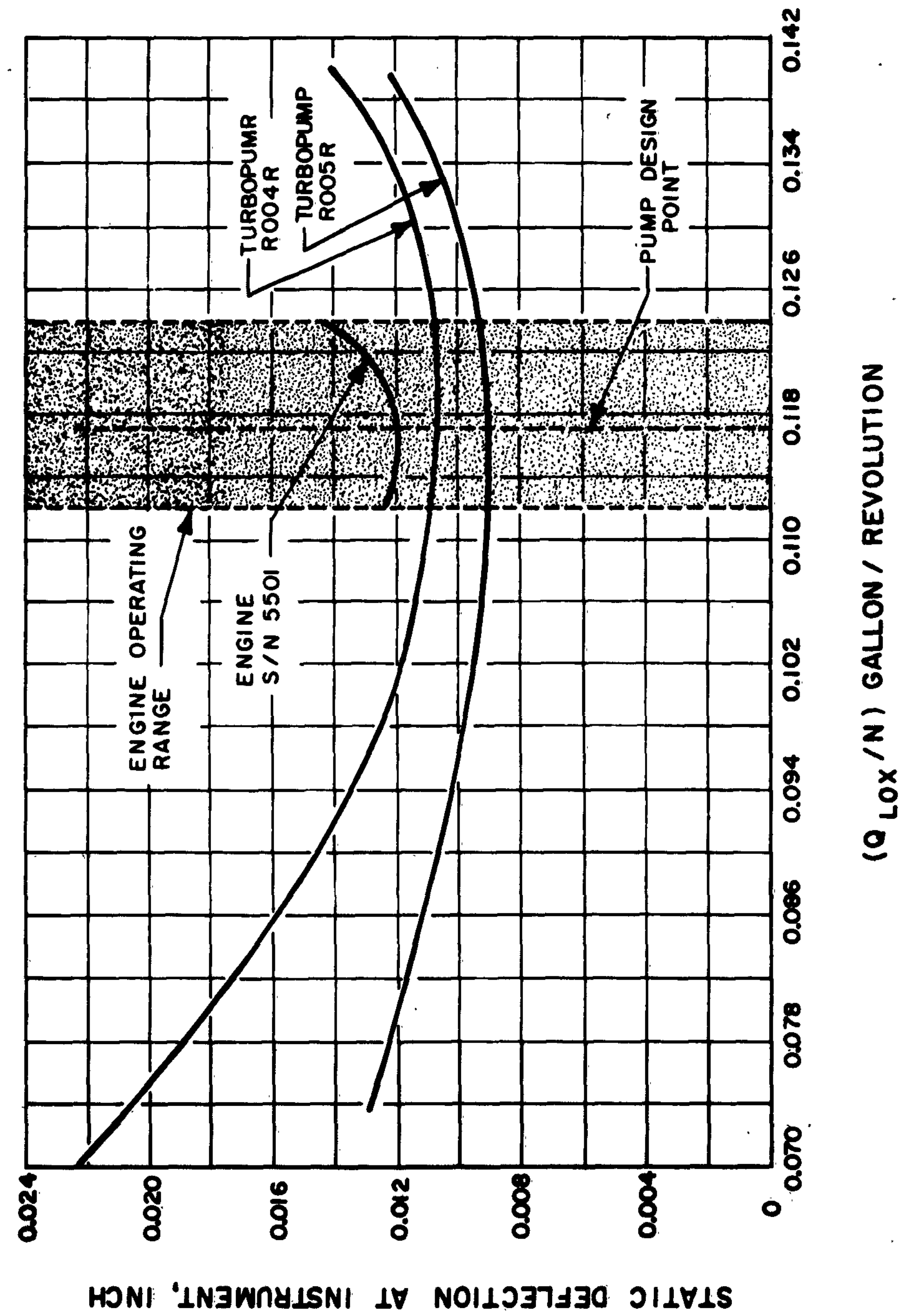


Figure 20. Static Deflection Characteristics

MIXTURE RATIO EFFECT ON STATIC SHAFT DEFLECTION AT WEAR RING			
Data Source	Static Deflection at -15% Mixture Ratio, inch	Static Deflection at Nominal Mixture Ratio, inch	Static Deflection at +15% Mixture Ratio, inch
Engine S/N 225501	0.0088	0.0082	0.0102
Pump S/N R004R	0.0075	0.0074	0.0075
Pump S/N R005R	0.0061	0.0060	0.0063
Average	0.0075	0.0072	0.0080

Figure 21 shows how varying the mixture ratio will affect vibration deflection, and reveals that the amplitude of the vibrations decreases as mixture ratio increases. The mean vibration level at nominal mixture ratio is 0.006 inch (center-to-peak) at the instrument, corresponding to a vibration of 0.0039 inch (center-to-peak) at the wear ring.

The mean vibration level increases to 0.007 inch (center-to-peak) at the instrument when mixture ratio decreases to -15%, and decreases to about 0.004 inch (center-to-peak) when the mixture ratio increases to +15% of nominal. An increase in pump speed increases the amplitude of the vibrations (Fig. 22); this is expected from theoretical considerations. As the mixture ratio (or oxidizer flowrate) increases, vibration amplitude decreases. Because of the increased oxidizer flowrate, the viscous damping, or friction effect, also increases. A graph showing the mean vibration amplitude at various oxidizer flowrates is shown in Fig. 23.

It should be noted that some question exists as to whether rubbing occurred during the mixture ratio excursion tests. Deflections were experienced which were of a magnitude large enough to cause rubbing within a minimum

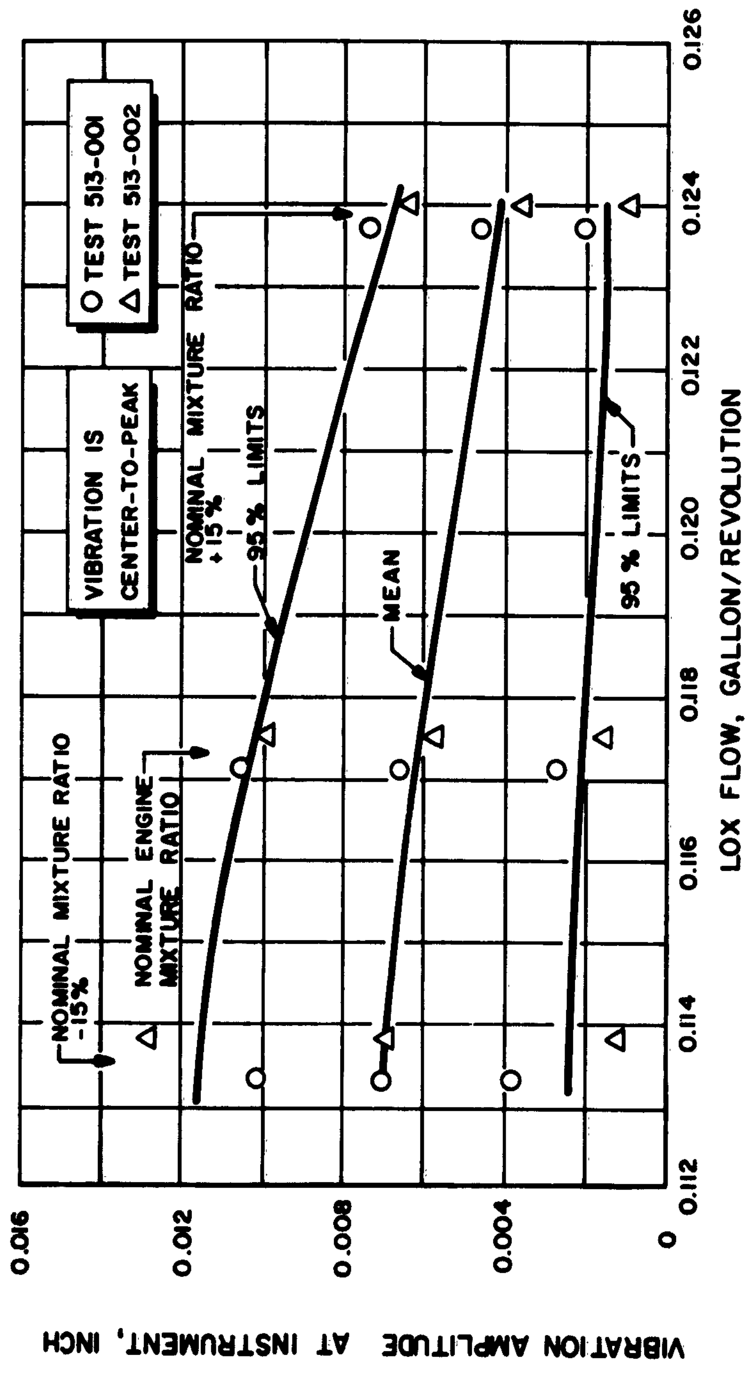


Figure 21. Effects of Mixture Ratio on Vibration

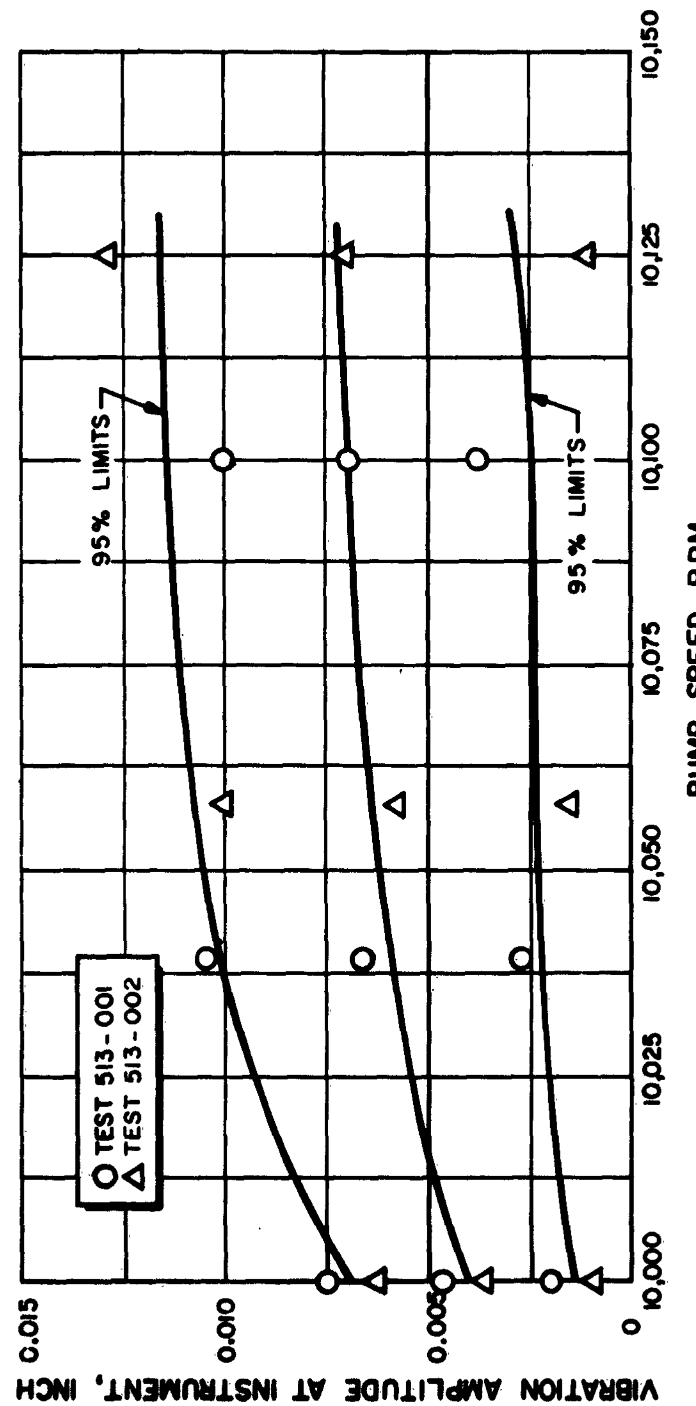


Figure 22. Effect of Pump Speed on Vibration

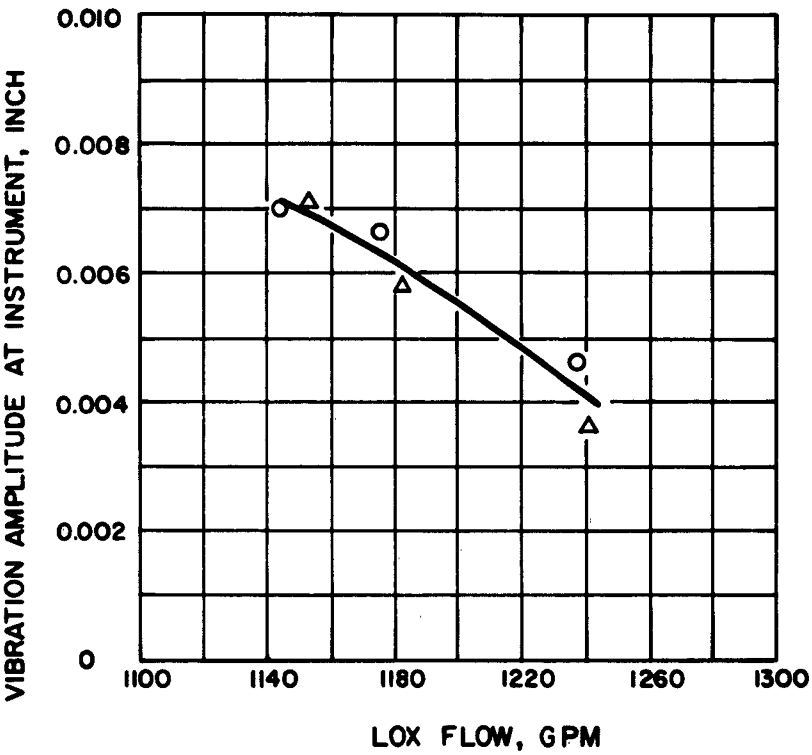


Figure 23. Effects of LOX Flow on Vibration

assembly clearance pump, but it was not known if the assembly clearances for this Kel-F liner approached the minimum clearances. Inspection of the test data revealed no conclusive evidence of rubbing, and rub marks were not reported on the Kel-F liner installed during the tests.

Valve Operating Rate Variations

Figure 24 shows the different valve operating rate variations that were studied during the engine test series. Evaluation of the shaft deflection data revealed no discernible effects caused by the variations.

Shaft Deflection Frequency

Figure 25 shows a sonic analysis photograph obtained from the magnetic tape recording instrumentation. This photograph represents the time period 4 to 6 seconds after ignition during test 513-006 on sustainer engine S/N 225501, and shows that the primary frequency of vibration was approximately 170 cps. This corresponds to the 170-cps frequency of rotation of the turbopump. The photograph displays no apparent harmonies, but shows that the wave form is not purely sinusoidal. A typical shaft deflection oscillograph recording was presented in Fig. 11; this recording also showed that the wave form is not a pure sinusoid.

Clearance Reduction Estimates

Table 14 presents the running clearance estimates for the Mark 4 turbopump with minimum clearance conditions and with nominal conditions.

The thermal effect clearance reductions result from chilling the pump assembly from ambient to LOX temperature. The clearance reduction due to concentrictiy has been eliminated by improving inspection techniques, but this reduction still exists an older turbopumps. Centrifugal effects are caused

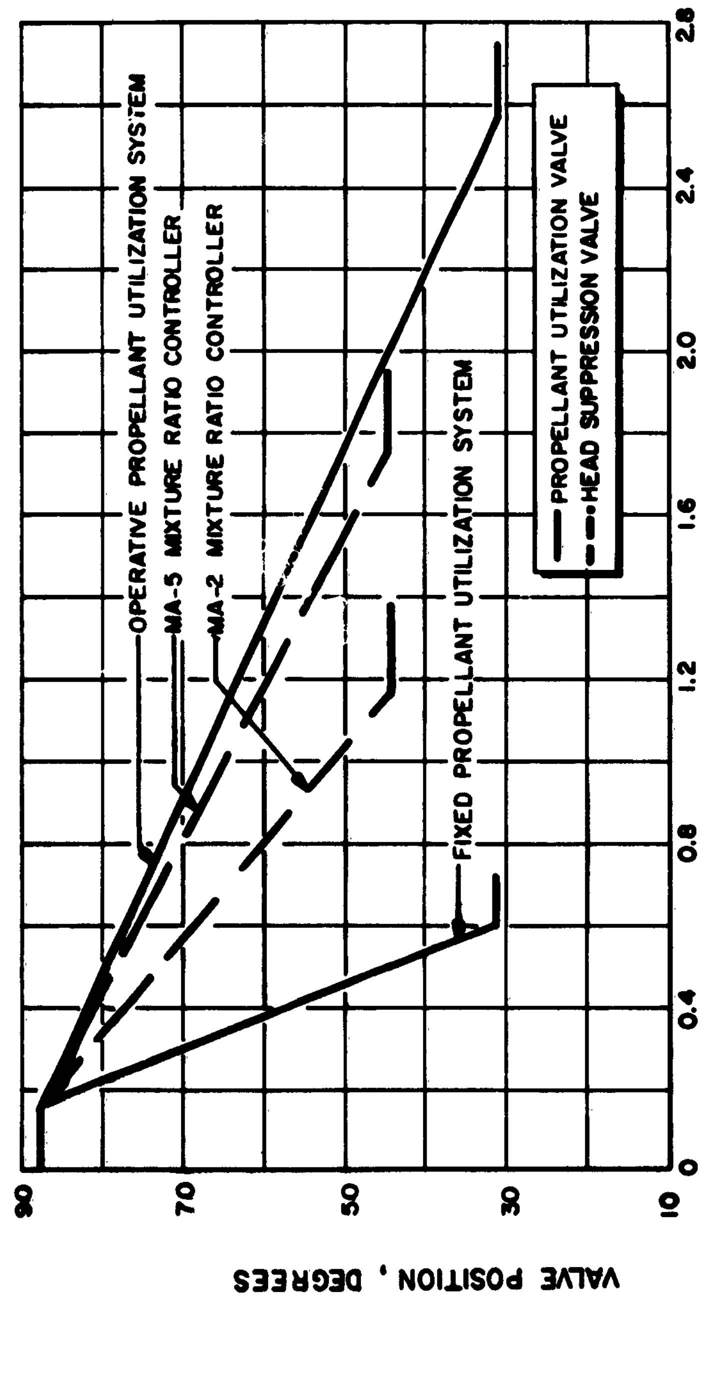
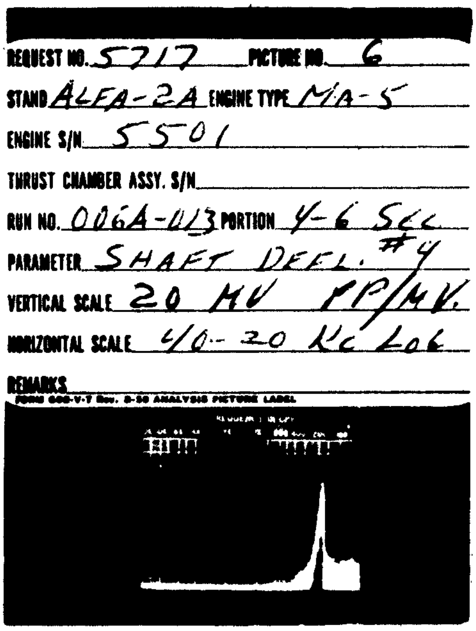


Figure 24. Valve Operating Rates

Amplitude



Frequency, cps

Figure 25. Shaft Deflection Frequency

TABLE 14

MARK 4 LOX PUMP CLEARANCE ESTIMATES

	Nominal Clearance, inch			Minimum Clearance, inch		
	Diverter Lip	Wear Ring	Kel-F Liner	Diverter Lip	Wear Ring	Kel-F Liner
Assembly Clearance	+0.0415	+0.0318	+0.0279	+0.040	+0.025	+0.023
Thermal Effect	-0.006	+0.004	-0.0035	-0.006	+0.004	-0.0035
Concentricity Effect	-0.003	- - -	- - -	-0.003	- - -	- - -
Centrifugal Effect	+0.0007	-0.0007	-0.0007	+0.0007	-0.0007	-0.0007
Missile Duct Loads Effect*	-0.0056	-0.006	-0.006	-0.0056	-0.006	-0.006
Maximum Running Deflection	-0.0250**	-0.0164	-0.0164	-0.0250**	-0.0164	-0.0164
NET RUNNING CLEARANCE	+0.0026	+0.0127	+0.0013	+0.0011	+0.0059	-0.0036

*Duct load effect based on tests conducted on MA-3 system

**Includes effect of shaft axial motion based on MA-3 tests (normal start) by strain resulting from inertia loads in the rotating impeller. Missile duct load clearance reductions are caused by misalignment of the low-pressure propellant feed ducts and forces introduced into the system by engine gimbaling. Shaft deflection clearance reductions are the result of the static deflection due to hydrodynamic forces and the vibration deflection. Shaft deflection clearance reductions were measured during the test series and the maximum observed deflections are presented in Table 14. The diverter lip clearance reduction figures contain the effect of axial movement of approximately 0.007 inch which was measured during solder cone tests described in Appendix D.

The initial clearances shown in the nominal clearance columns of Table 14 were determined from an average of all actual turbopump buildup measurements as of March 1963. The nominal clearance values, based on data from the two engines tested, show that clearances will exist at all times within the nominal pump during normal start and mainstage operation. The minimum clearance values show that, even if the initial assembly clearance is minimum and all clearance reductions are maximum and in the same direction, running clearances will exist in the pumps with metal wear rings. Under these conditions, interference of 0.0036 inch will exist within the Kel-F lined pumps.

The above possible interference condition is not considered detrimental, based on the following summary of Rocketdyne rubbing experience with Kel-F in turbopumps. Twenty-two previously described rubbing tests were conducted, and 20 of these were solid-propellant gas generator firing tests where mainstage operation was not achieved. These 20 tests were performed with an MA-3 series sustainer engine. The additional two tests were conducted with an MA-5 series engine, and a rubbing duration of 21 seconds was accumulated during mainstage. A maximum rub depth of 0.009 inch was observed during an MA-3 test that involved a delayed opening of the head suppression valve.

This rub was purposely caused. No difficulties or adverse effects were encountered during any of the rubbing tests. Component rubbing tests amounted to nine tests for a total duration of 106 seconds with no adverse effects.

As a matter for comparison, the Mark 15 LOX pump is designed so that rubbing between metal and Kel-F occurs at three places in the pump approximately 75% of the time. These three contact locations are: (1) the inducer rubbing against the Kel-F inducer shroud, (2) the impeller forward labyrinth seal (Kel-F) rubbing against the impeller, and (3) the impeller aft labyrinth seal (Kel-F) rubbing against the impeller. The Mark 15 inducer blades normally cut grooves of approximately 0.005 to 0.007 inch depth in the Kel-F inducer shroud. The Mark 15 LOX pumps have experienced more than 180 tests and 25,000 seconds of accumulated duration with no failures or deleterious effects attributed to rubbing. The Atlas R&D engines have accumulated approximately 34,000 seconds of test time with Kel-F liners installed in the oxidizer pump inlet adapters.

Effects on Deflection Phase Angle

As the pump operating point varies, the phase angle of the deflection will rotate. Figure 26 shows the phase angle orientation for approximately nominal mixture ratio (0.117 gallon of oxidizer flow/revolution), and Fig. 27 shows a plot of phase angle vs oxidizer flow/revolution. During engine tests, the deflection was oriented at approximately 30 degrees for nominal mixture ratio. It was found that the phase angle rotated clockwise as the mixture ratio was increased. Figure 26 shows that the phase angle is sensitive at an oxidizer flow/revolution close to the design point. The solid line encompasses all component and engine test data points, and the dashed line is the shape of the curve that would be predicted by theory (Ref. 2).

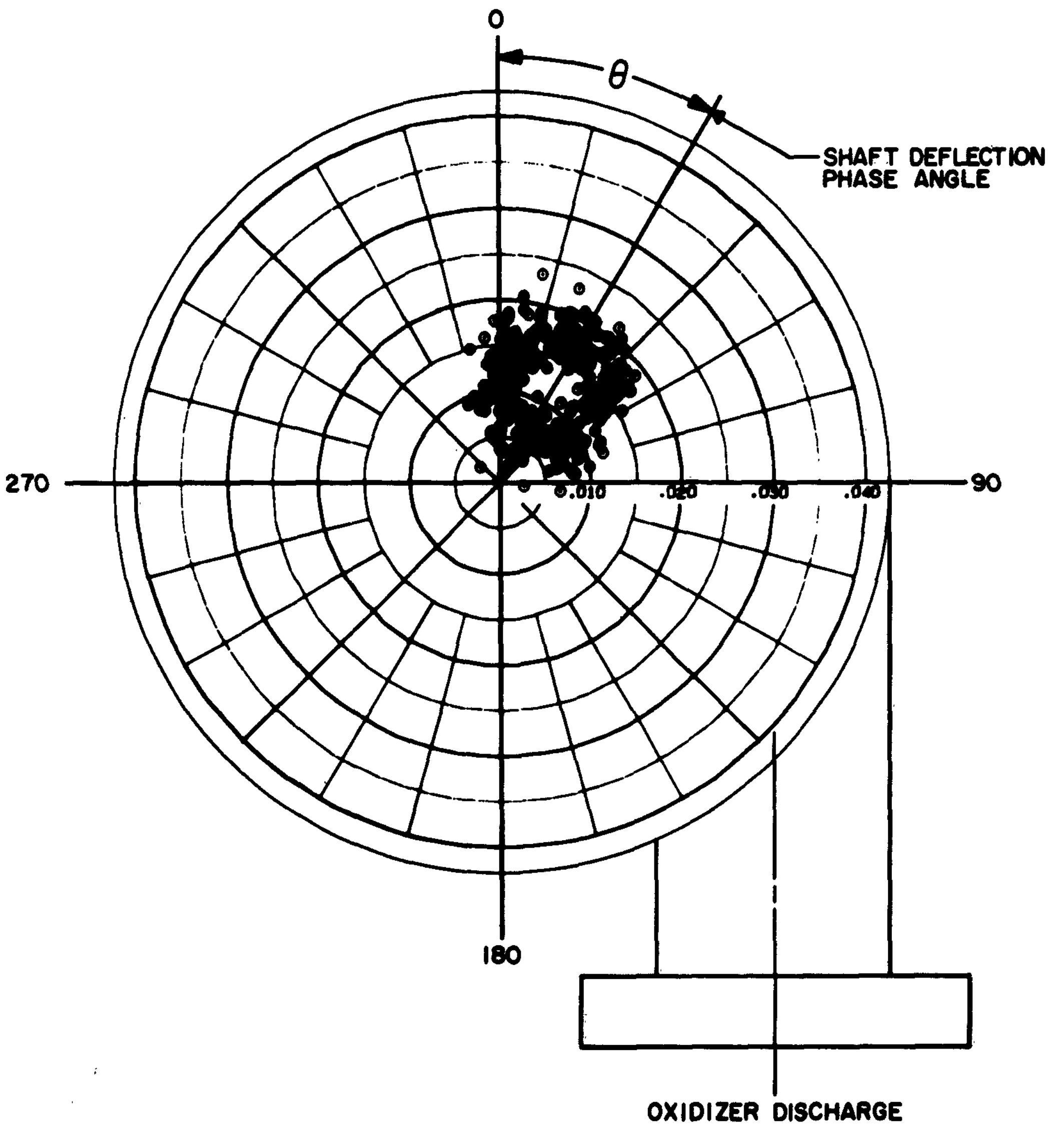


Figure 26. Shaft Deflection Phase Angle Orientation

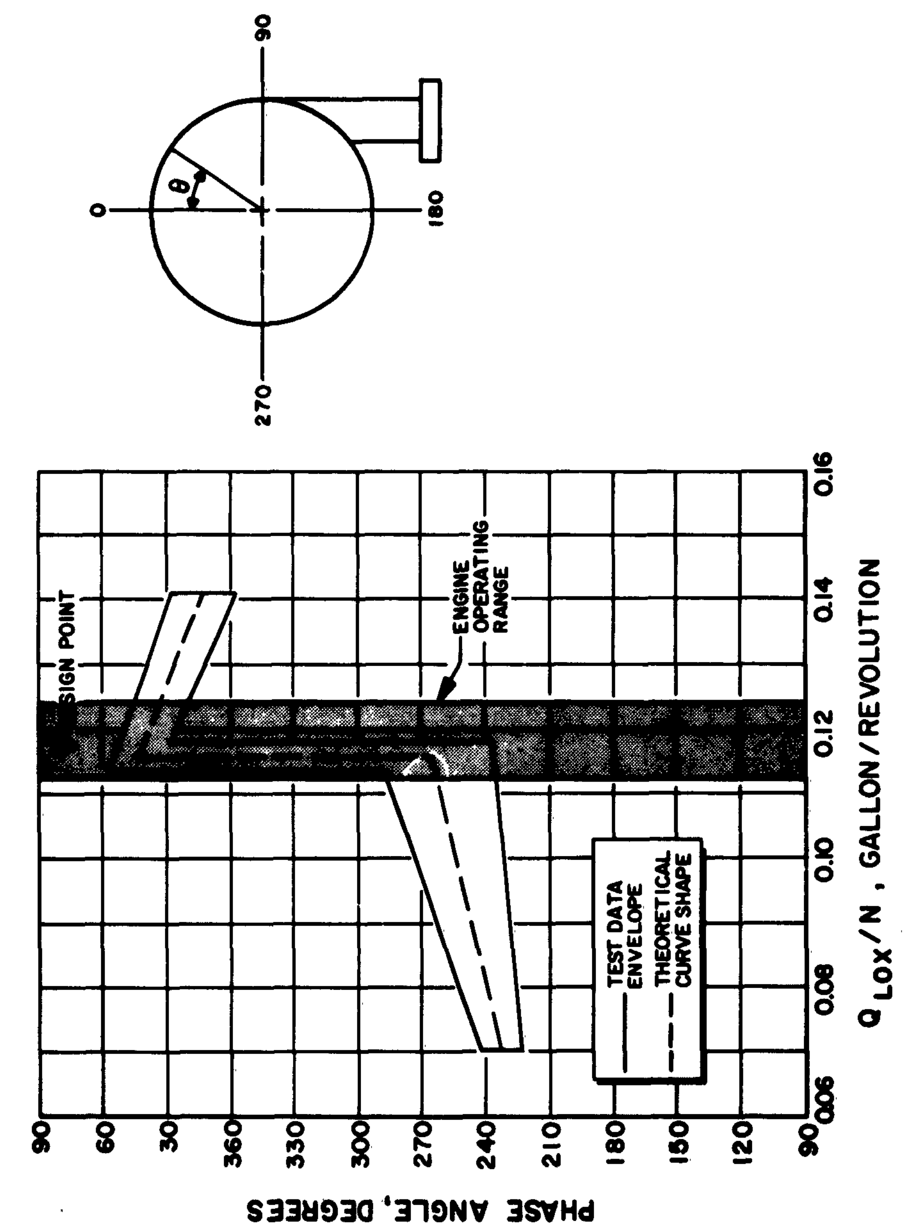


Figure 27. Deflection Phase Angle Characteristics

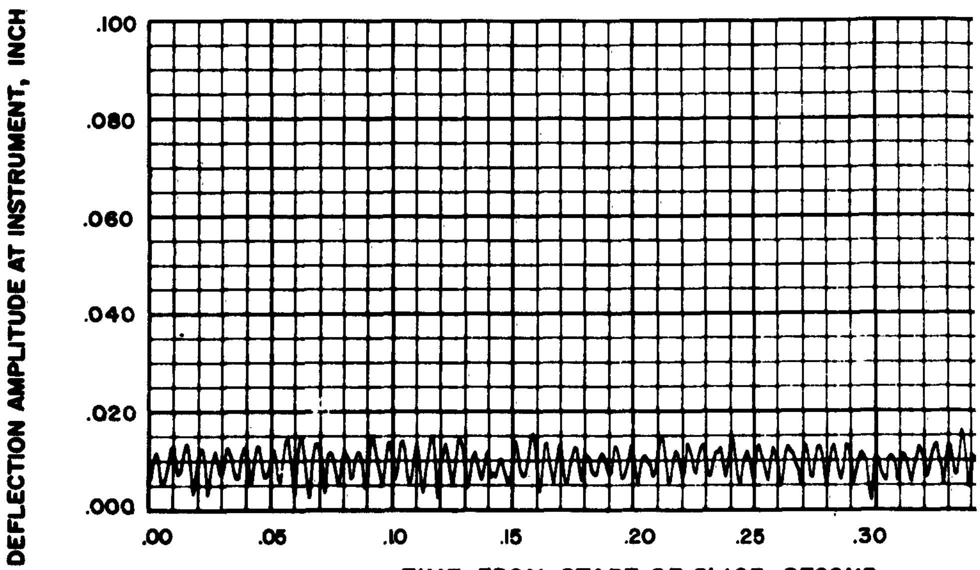
Effects of Rubbing on Shaft Deflection

As explained in the engine tests portion of this report, rubbing was induced during one of the two tests accomplished during the shafts deflection test series. It was not positively concluded that rubbing had occurred on the other test. Figure 28 presents a comparison of digital data obtained from test 513-006, where rubbing was not attempted and test 513-008, where rubbing was positively induced. Vibration amplitudes of these tests show that rubbing between the impeller and the Kel-F liner has the effect of reducing the amplitude of vibration. A comparison of the polar plots of the turbopump shaft position (Fig. 28) shows that the steady-state deflection of the shaft was considerably reduced, or shifted, during the test where rubbing occurred. In this figure, the steady-state deflection differs by approximately 0.008 to 0.010 inch between the two tests. These tests were conducted under the same conditions and the data slices were taken at approximately the same time.

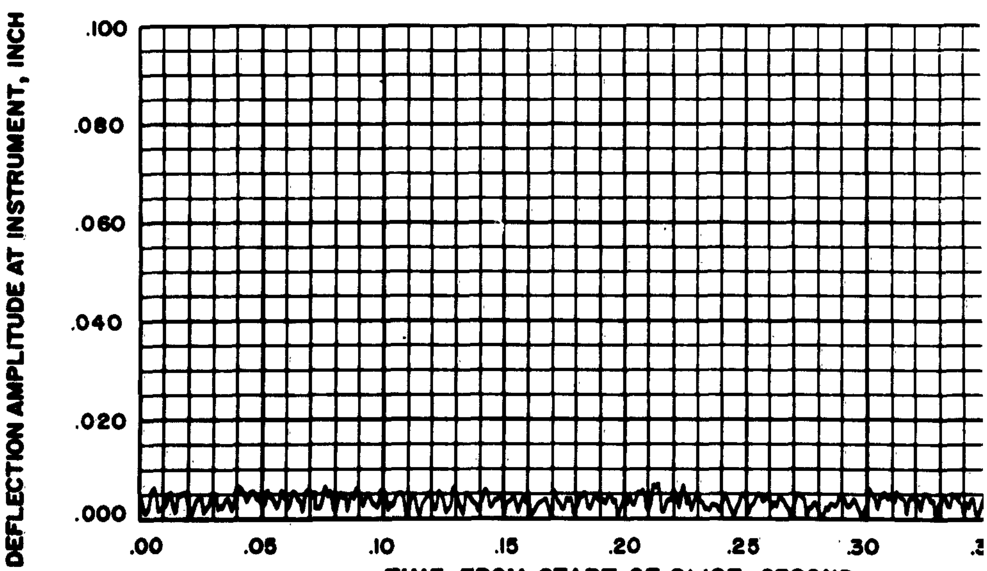
COMPONENT TEST RESULTS

Oxidizer Pump Operating Point Variations

Figure 29 shows the pressure tap locations on the fuel pump. Figure 30 shows the fuel pump radial pressure distribution for various flowrates and reveals that the pressure distribution in the fuel pump is fairly even immediately downstream of the volute tongue, but as the scroll area increases there is a slight dip in pressure in the volute at about 170 degrees downstream from the volute tongue. A steady pressure rise occurs from this point. This is evident for all flowrates, with the maximum differential pressure of 240 psi occurring at the lowest flow of 180 gpm. Figure 31 shows the fuel pump radial shaft loading for increasing flowrate. At the

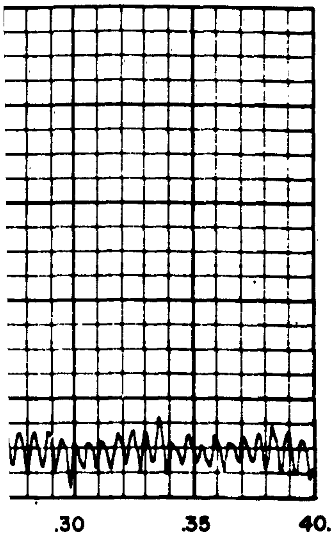


Shaft Deflection With No Rubbing, Test 513-006



Shaft Deflection With Rubbing, Test 513-008

1



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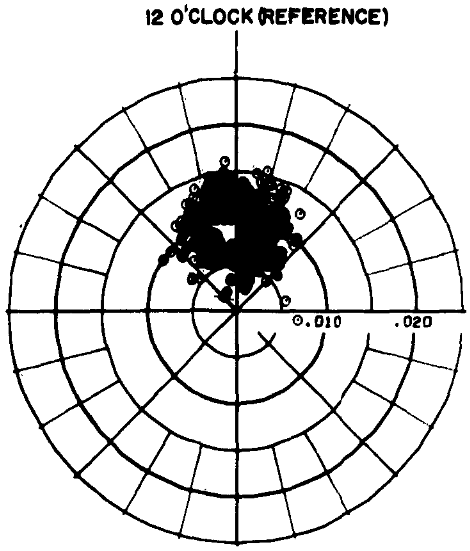
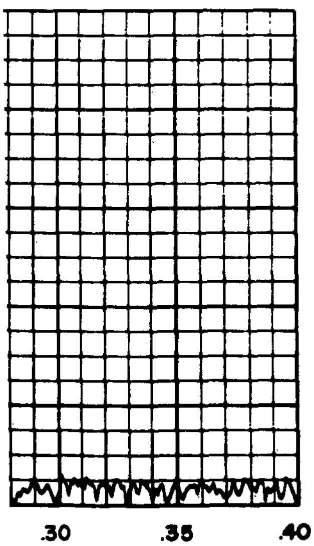


Turbopump Shaft Position With No Rubbing, Test 513-006



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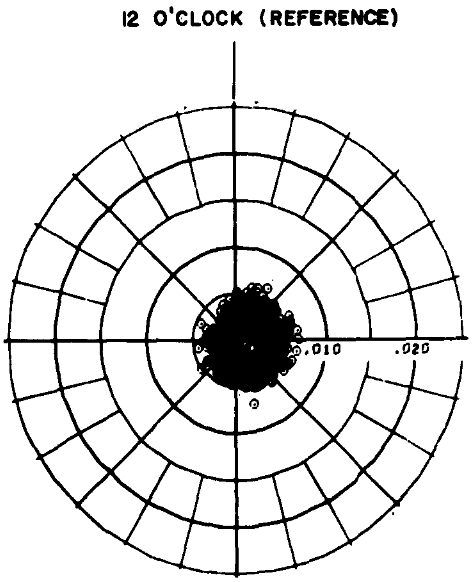
513-008



Turbopump Shaft Position With Rubbing, Test 513-008

Figure 28. Comparison of Shaft Deflection During Test 513-006 and Test 513-008

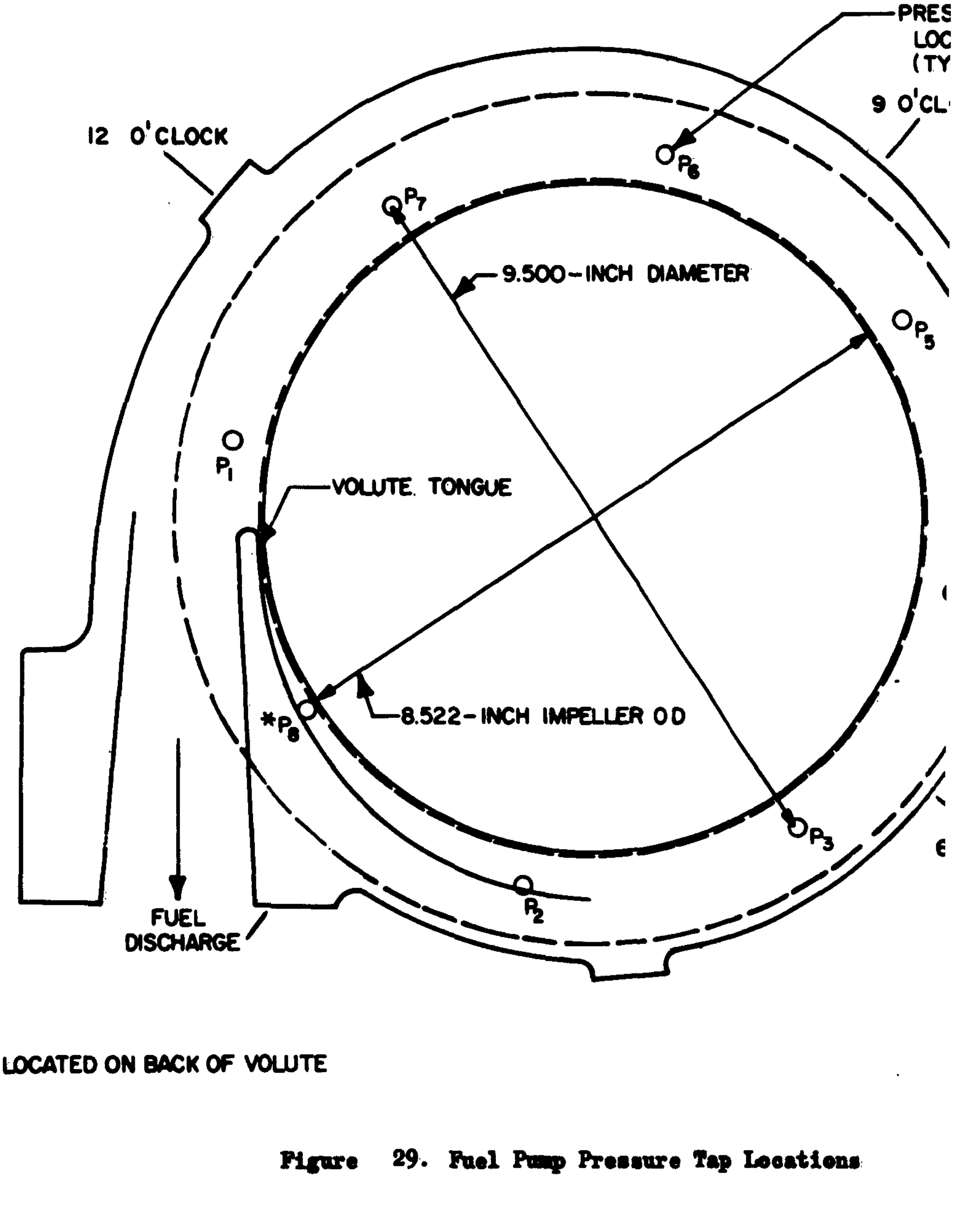


Figure 29. Fuel Pump Pressure Tap Locations

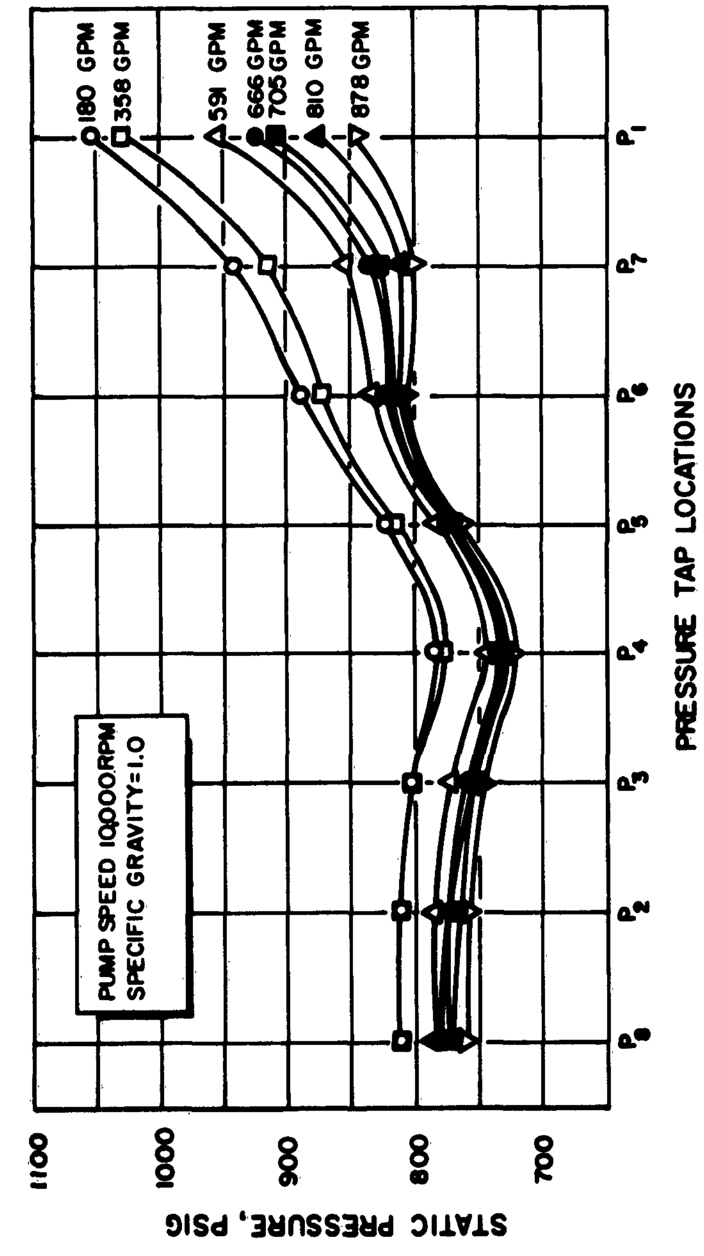


Figure 30. Fuel Pump Radial Pressure Distribution

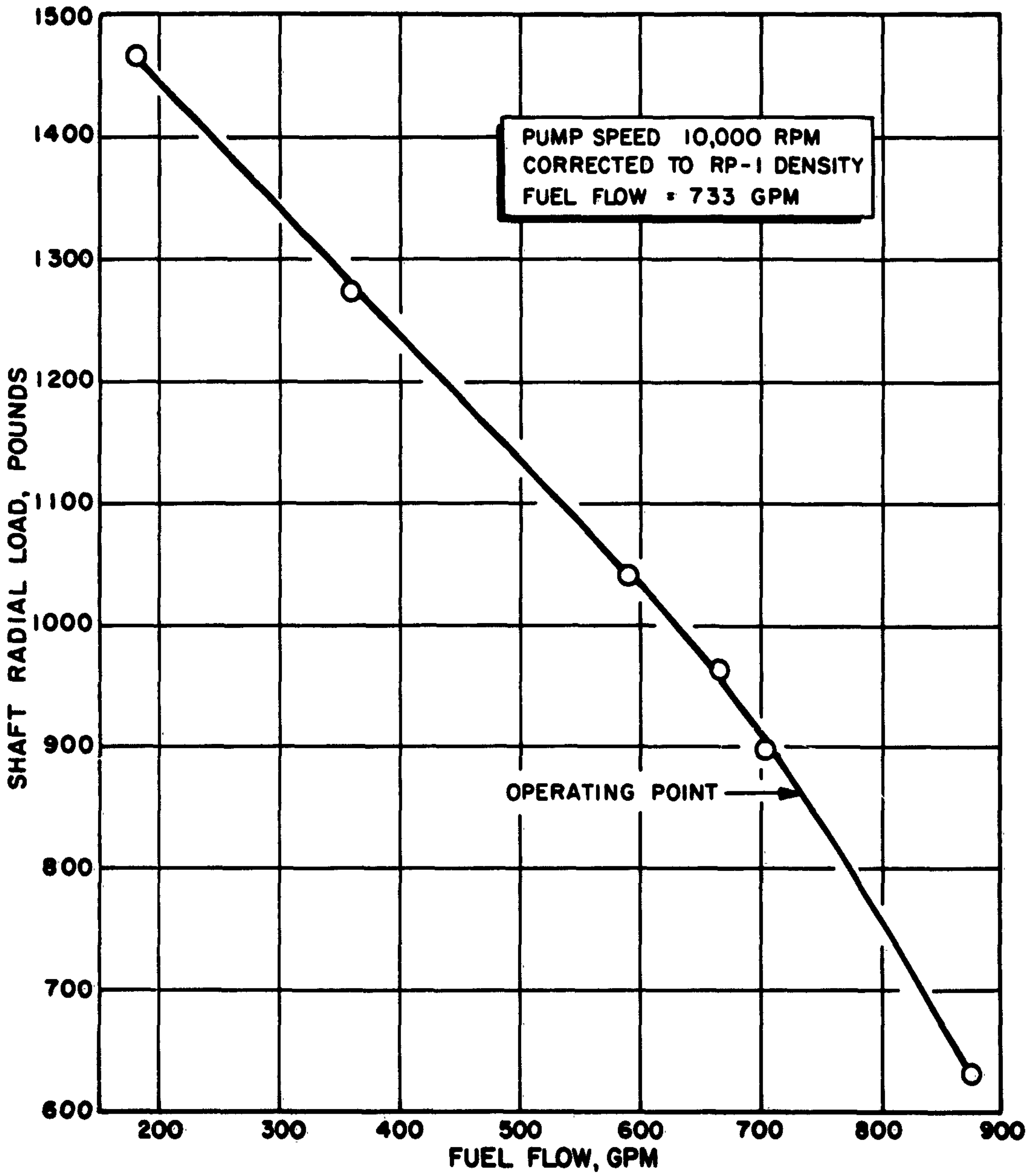


Figure 31. Fuel Pump Shaft Radial Loading

fuel pump design flow (733 gpm), the load is approximately 865 pounds (corrected for RP-1 density). The test speed was 10,000 rpm. Data obtained from the fuel pump pressure distribution indicate that the fuel pump deflection for all test flows is directed toward 6:00 to 6:15 o'clock.

Analysis of these tests has shown that variation of the fuel pump operating point over the normal sustainer engine operating range can be expected to influence the deflection of the pump shaft at the LOX pump wear ring by less than 0.0005 inch.

Oxidizer Pump Cavitation

Several cavitation tests were made using both LOX and liquid nitrogen as the test fluid. The static deflections for the cavitation tests are plotted in Fig. 32. The magnitudes of both the static and vibration deflections did not appear to change appreciably during cavitation, but a low-frequency oscillation was introduced which increased the total shaft deflection. Figure 33 shows a portion of an analog record of shaft deflection data taken immediately prior to and during cavitation.

Air and Hydrodynamic Pressure Tests

Figure 34 shows the location of the pressure taps in the LOX pump volute for the air tests. Figure 35 shows the pressure distribution in the LOX pump for various flowrates. At flowrates above design (1178 gpm), radial pressure distribution around the LOX pump volute was reasonably even. At design flow and below, the pressure distribution was still fairly even immediately downstream of the volute tongue; however, as the scroll area increases, there is a rise in volute pressure. The maximum differential

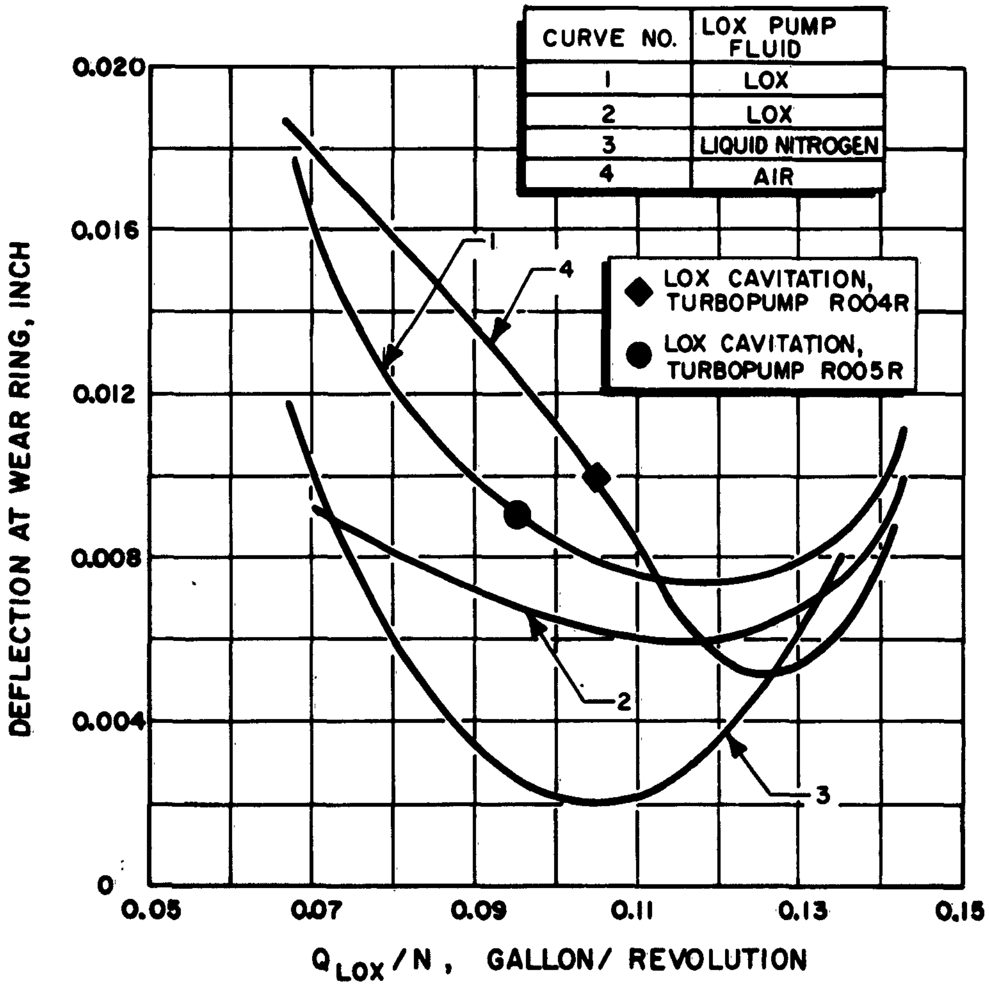


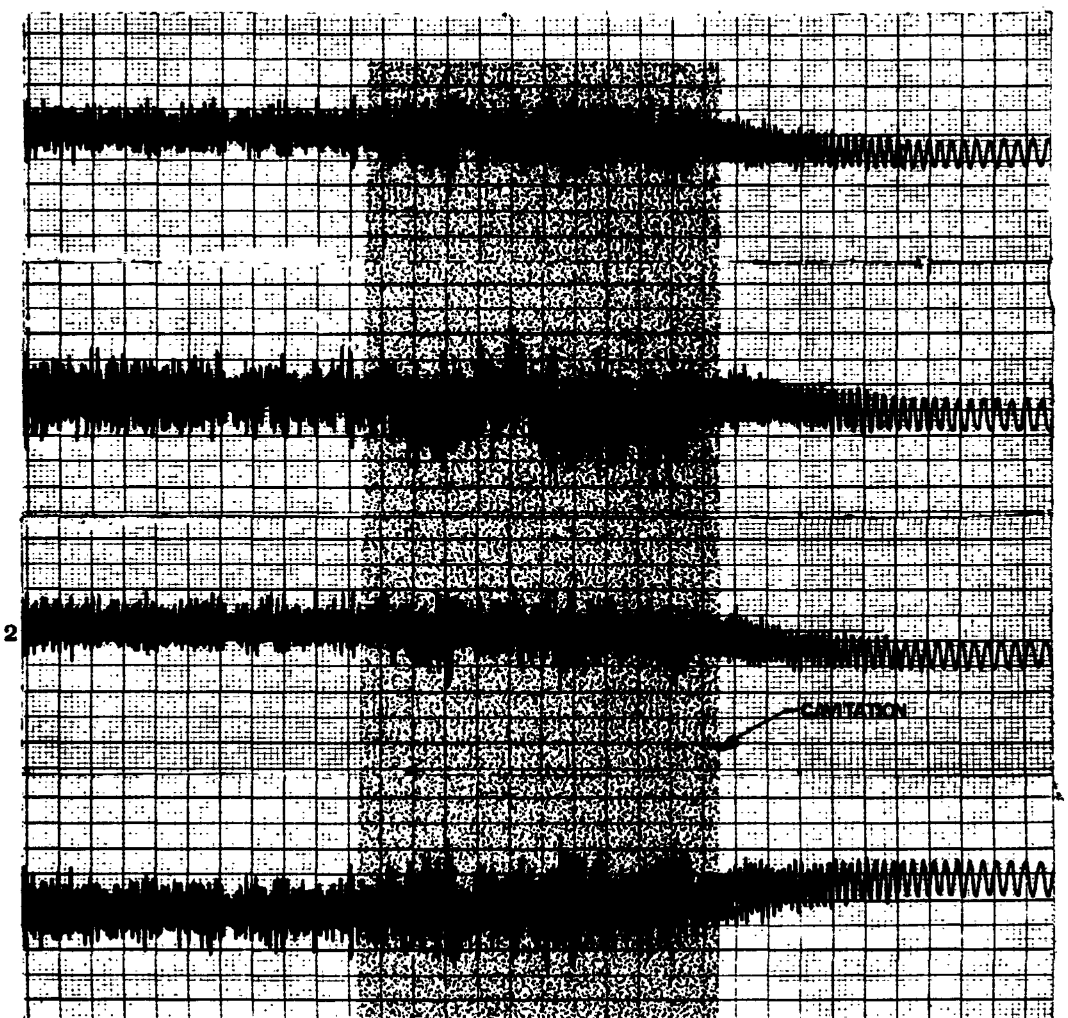
Figure 32. Static Deflection Characteristics

Transducer No. 1

Transducer No. 2

Vector Summation
of
Transducer No. 1 and 2

Transducer No. 3



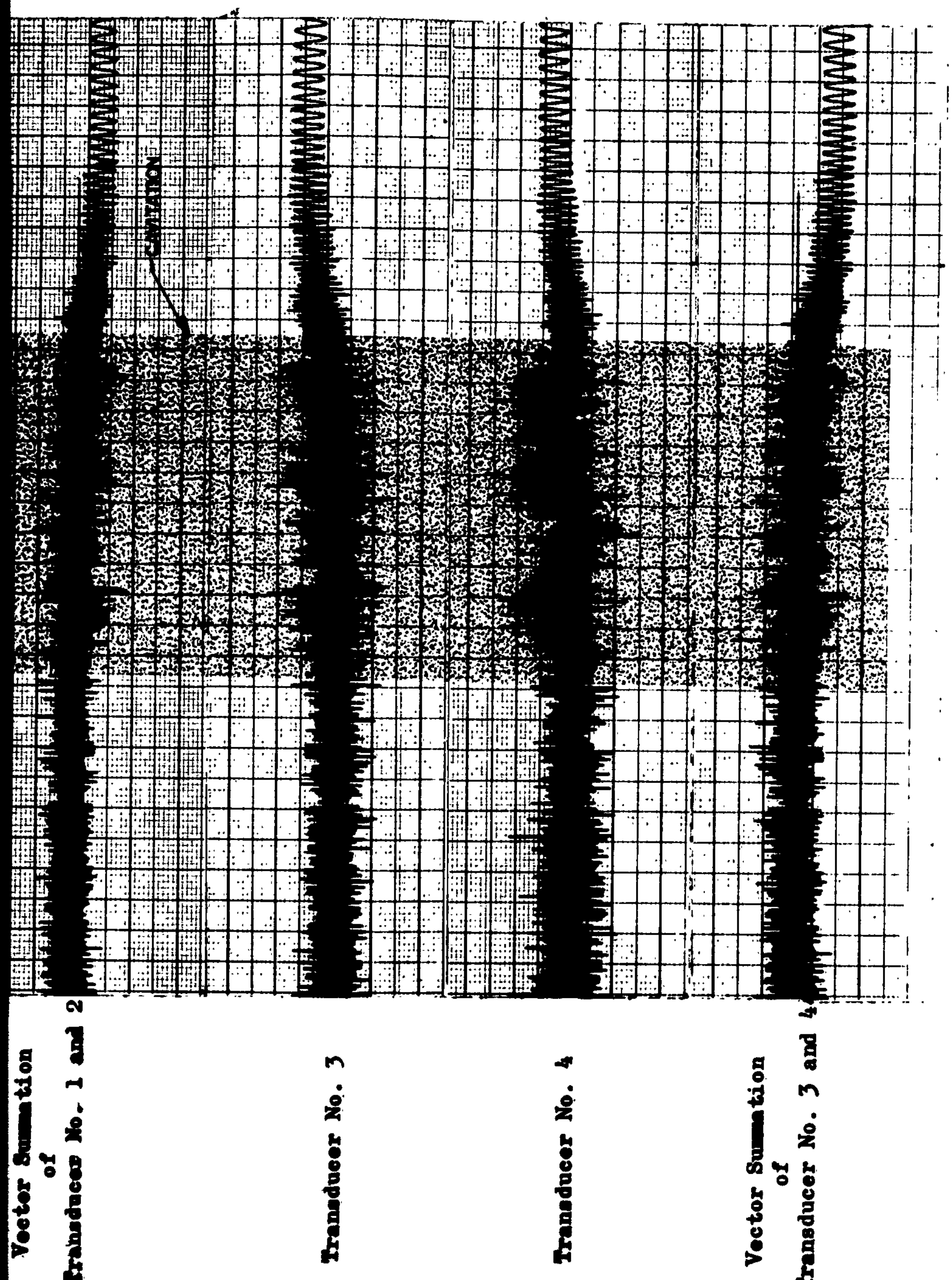


Figure 33. Cavitat
Shaft D

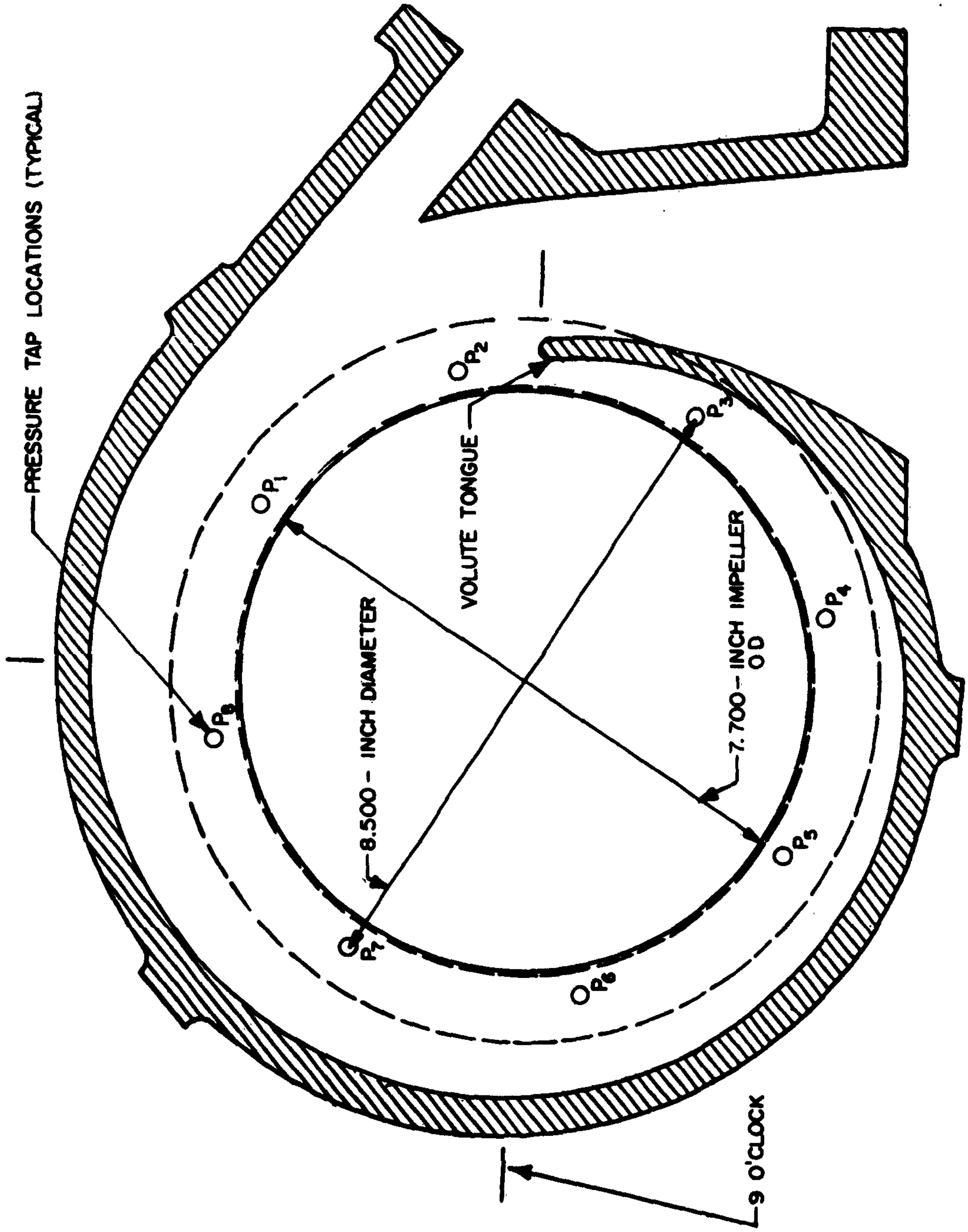


Figure 34. LOX Pump Pressure Tap Locations

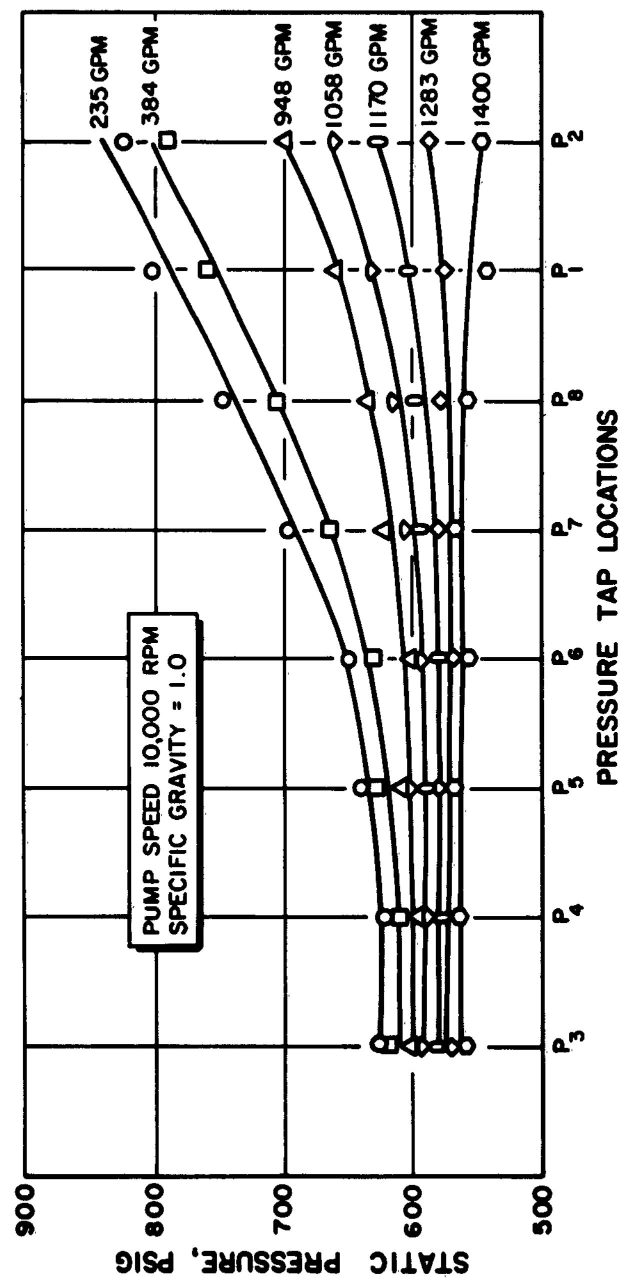


Figure 35. LOX Pump Radial Pressure Distribution

pressure measured from the volute tongue to the scroll area immediately downstream from the discharge cone was 210 psi at a flowrate of 235 gpm. Shaft radial loading at LOX pump design flow, computed from the above radial pressure distribution data, is approximately 200 pounds (corrected for LOX density).

Using a LOX volute 12:00 o'clock position as a reference, the shaft deflections in the LOX pump were calculated to be in the direction of about 7:15 o'clock for 80 to 110% of the design flow. At 120% design flow, the shaft deflection was calculated to move almost 180 degrees to approximately the 1:30 o'clock position. This transition was found to occur over a very slight change in flow. These data and conclusions correlate closely with engine and component shaft deflection test results. The loads calculated from pressure distribution data are the static component of shaft deflection only and do not represent the vibration effect.

Alternate Fluid (Liquid Nitrogen) Operation

Figure 32 compares the magnitude of the LOX pump steady-state shaft deflections at various flowrates and using various test fluids. LOX was used to obtain curves 1 and 2, liquid nitrogen was used to obtain curve 3, and air was used and corrected to LOX density to obtain curve 4. The liquid nitrogen used for curve 3 was not density-corrected so that the density effect could be distinguished. Curves 1 and 2 were derived from component test data and represent shaft deflections actually measured by the shaft deflection measuring instrument. Curve 1 represents turbopump S/N R004R, curve 2 represents turbopump S/N R005R, curve 3 represents turbopump S/N R004R, and curve 4 represents the static shaft deflection calculated from pressure-distirbution data from the air tests.

Water was used as the test fluid in the fuel pump for these tests and the curves were plotted for approximately constant fuel pump flow/revolution. These data demonstrate a variation in the steady-state deflection between turbopumps and give evidence that the steady-state deflection varies with the density of the fluid being pumped. The data indicate that the deflection was decreased by replacing LOX with liquid nitrogen. Analysis was complicated by the fact that there were variations between similar runs on the same turbopump. Liquid nitrogen is less dense and less viscous than LOX at a given temperature, and it would be expected that the fluid with the lesser density would exert less hydrodynamic loading and, therefore, cause less deflection.

This hypothesis is backed by data taken during the air tests. Very little differential pressure is observed from the tongue of the volute to the discharge section of the pump when air is used as the test fluid. From this observation, it would be expected that the shaft hydrodynamic loading would be very little and, thereby, the shaft deflection would be very low. These data, when corrected to LOX density, show a much higher pressure distribution and a deflection curve quite similar to those of pumps using LOX as the test fluid. This correlation can be seen by comparing deflection curves 1, 2, and 4 of Fig. 32.

Comparison of Shaft Deflection and Solder Cone Tests

A comparison of LOX pump shaft deflections during tests with solder cones (Appendix D) and deflections observed during the MA-3 Kel-F liner interference tests is outlined in subsequent paragraphs.

Results of tests with solder cones on the inlet adapter wear ring indicated that a clearance reduction in the range of 0.034 to 0.036 inch could be expected to be experienced when the pumps were started and the head suppression valve was left closed. Because these tests were, of necessity, conducted with liquid nitrogen as the test fluid in the oxidizer pump, the observed deflections are considered to be slightly less than would be expected had the test fluid been LOX.

This increase in magnitude of shaft deflections when LOX is used as the test fluid can be deduced when consideration is given to the factors which cause shaft deflections. These factors are (1) internal hydrodynamic forces acting on the fuel and oxidizer impellers, (2) internal forces acting on the drive gear, (3) the critical speed of the shaft and associated masses, and (4) the unbalance of the rotating assembly. When extrapolating results obtained with liquid nitrogen to service with LOX, all these forces can be assumed to be constant except those internal forces which act on the oxidizer impeller.

The resultant hydrodynamic force on the oxidizer impeller at design point turbopump operation has been found to be small. However, when the turbopump is operated at other than design point, such as when it is pumping against closed valves, a pressure gradient is developed around the periphery of the oxidizer volute. This pressure gradient manifests itself in a resultant radial load on the oxidizer pump impeller. The magnitude of this load is, of course, the factor which determines the radial displacement of the pump shaft from its no-load position.

Because the specific weight of LOX is approximately 14% greater than the specific weight of liquid nitrogen, it is estimated that the 0.034- to 0.036-inch shaft deflections observed in liquid nitrogen would be approximately equivalent to 0.039- to 0.041-inch deflections if the solder cone test fluid had been LOX instead of liquid nitrogen.

Upon examining the results of the MA-3 interference tests, it was found that when the Kel-F liner was offset so that it touched the impeller in the area of expected rubbing, the maximum rub depth observed when start sequences were initiated with closed head suppression valve was 0.009 inch. Because these tests were conducted with LOX as the test fluid, rub depths of approximately 0.039 to 0.041 inch would have been expected.

Tests on Kel-F buttons have shown that Kel-F presents a certain resistance to abrasion. It has been shown during ignition-stage only tests that Kel-F does not wear away quickly enough to permit accurate measurement of short-duration deflections. Because the Kel-F liner tests where rubs were experienced were of short duration, it is felt that indications of deeper rubbing and larger shaft deflections were not experienced because of two factors:

1. Kel-F resistance to abrasion
2. The elastic properties of Kel-F which present a counter load to the deflecting shaft and prevent it from deflecting as far as it normally would under the given conditions.

Based on these factors, it is concluded that the depths of the rub marks that occurred during the MA-3 interference test series are not indicative of the magnitude to which the shaft was attempting to deflect. It can also be concluded from this discussion and the previous discussion of the Mark 15 LOX pump rubbing experience that severe rubbing between the Kel-F liner and the impeller or inducer will not cause an explosion or any other deleterious effect.

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1. R-3498, Final Report, Program of Testing Nonmetallic Materials at Cryogenic Temperatures, Rocketdyne, a Division of North American Aviation, Inc., Canoga Park, California, 30 December 1962.

2. Holowenko, A. R.: Dynamics of Machinery, John Wiley and Sons, Inc., New York, 1955.

APPENDIX A

MATERIALS EVALUATION

A series of tests was performed to demonstrate the effectiveness of Kel-F when exposed to compressive stress. The primary purpose of these tests was to determine the long-term stress relaxation of heavy sections of Kel-F in compression. The results were used to predict the tensile load remaining in the bolts used to attach the Kel-F liners to the turbopump inlet housing after long periods of time at elevated temperatures.

Twelve test cells were designed and fabricated for this program. Each cell had four temperature-compensating strain gages attached to a hollow, thin-wall metal cylinder, and precision-ground pressure plates were fitted into each end of the cylinder. The bottom pressure plate on each cylinder was flat so as to provide maximum contact with the top of the test sample.

Load was applied to the cell by torquing a machine screw against a ball bearing that fitted into a socket in the top pressure plate. A rigid frame held the loading screw and supported the base of the test sample. Figure A-1 shows a sketch of a test cell. Calibration of the cells in compression and at various temperatures was accomplished using an Instron universal test machine and its temperature-controlled environment chamber. Load amplification and recording equipment were then calibrated against the test cells. A multichannel recorder was used to obtain the load decay data.

The test samples were loosely positioned in the cells and brought to test temperature before the desired load was applied. Load decay was automatically plotted on the 12-point recorder. The test cells used during the program do not provide the temperature compensation afforded by the Invar bolt spacers used in the actual application.

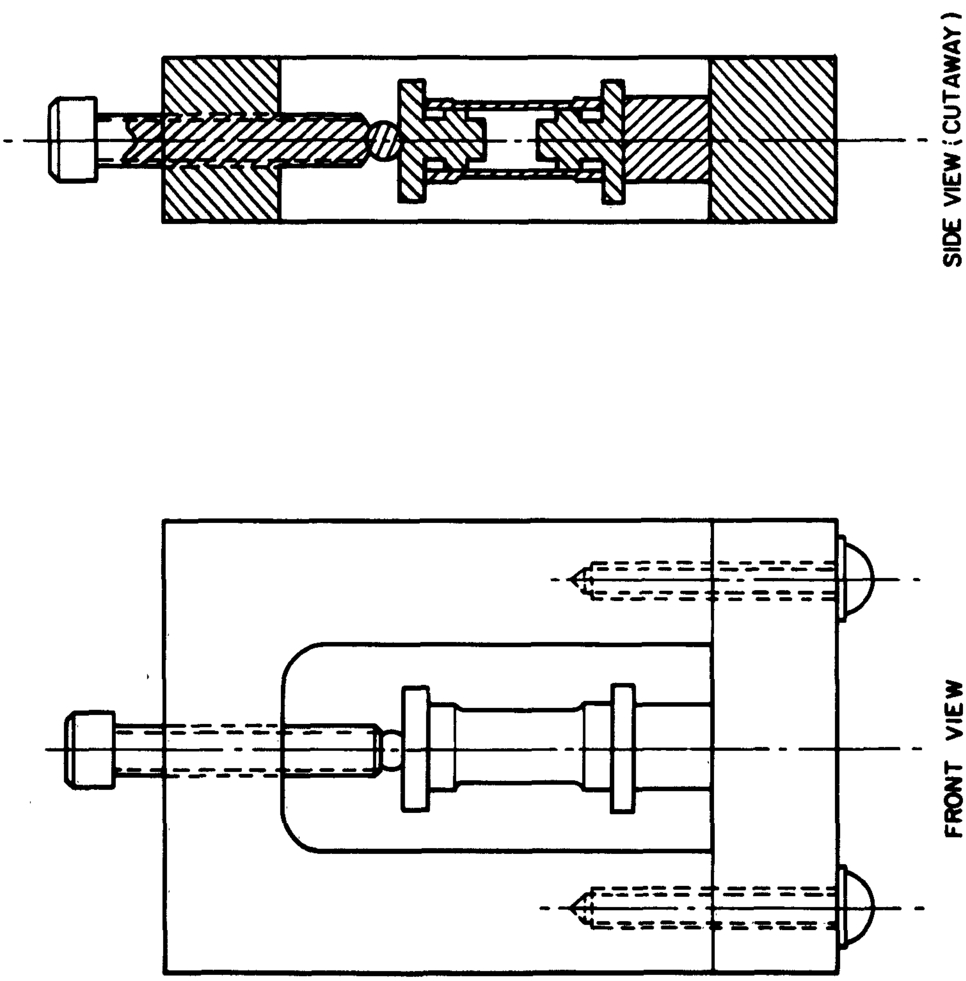


Figure A-1. Load Decay Test Fixture

As a part of the program, the effect of variations in initial load, sample length-to-diameter ratio, and test temperature were investigated to quantitatively determine their effect on the resistance of Kel-F to compressive stress relaxation. These tests continued for 7 days and resulted in a family of curves for each variable. The parameters investigated were:

1. Initial loads of 350, 500, 1300, and 1500 psi
2. Constant temperatures of 100, 120, and 160 F
3. Length-to-diameter ratios of 0.60, 1.00, and 2.00

With test procedures and initial loads approximating those used during turbopump buildup, the compressive force in the Kel-F sample had declined from the 350-psi initial load to 250 psi after 1400 hours at 160 F. Extrapolation of the data indicates that after 1 year (8760 hours) at a temperature of 160 F, more than 230 psi of compressive load would still remain in the Kel-F. This value is within the allowable limits of compressive load established for the Mark 4 LOX pump. One year at 160 F is more severe than the accumulated elevated temperature environment experienced by any turbopump. Currently, there is no way to extrapolate these data to other constant storage temperatures or to variable storage temperature conditions.

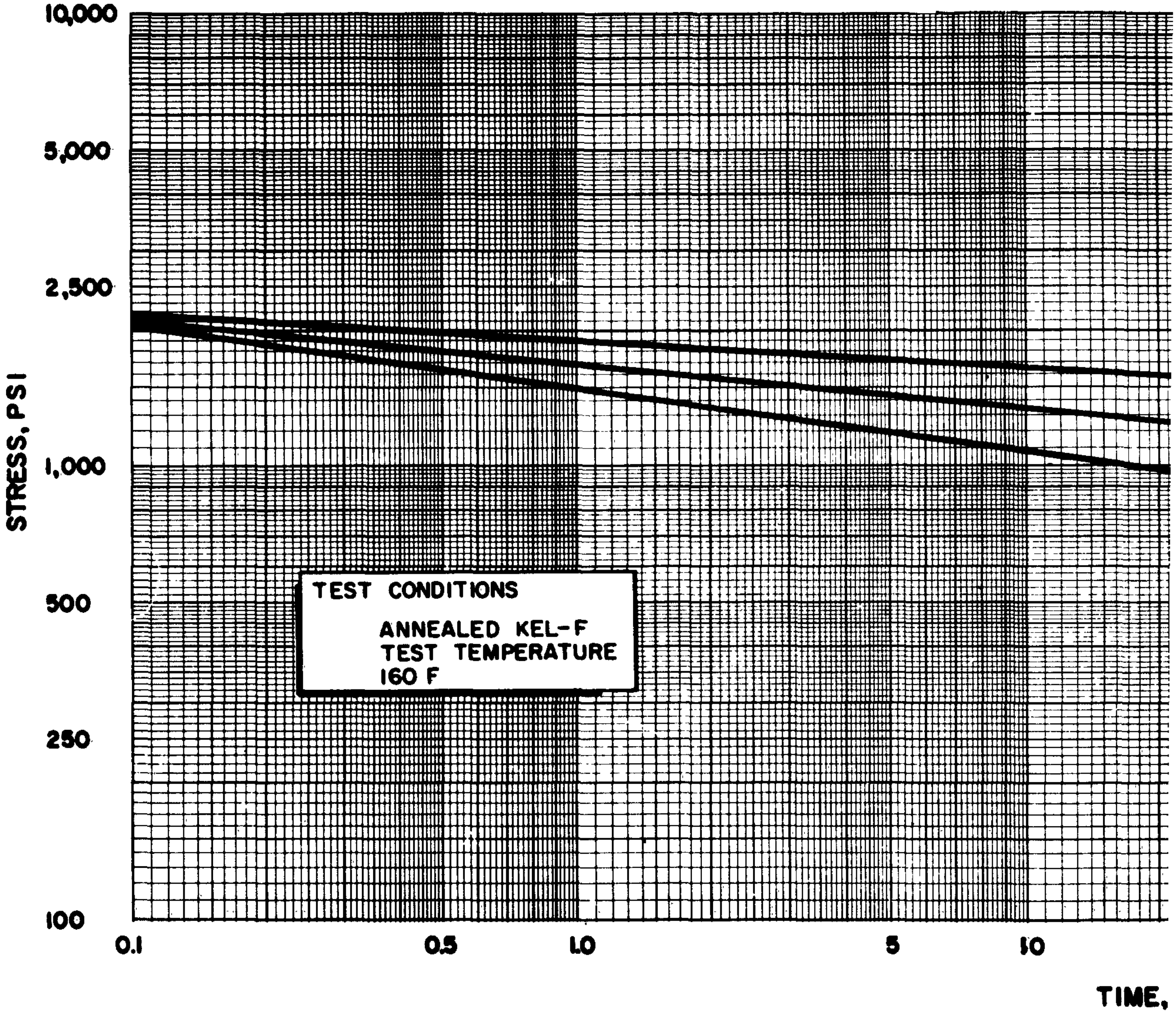
The greatest resistance to stress relaxation was found in short samples (small length-to-diameter ratio) at the lower test temperatures and lower initial loadings. Data for these tests are included as Fig. A-2 through A-4. The results of these tests were essentially as predicted from prior experience.

An interesting and unexpected property of Kel-F was found during this phase of the investigations. A repeat of one series of tests was performed using the same samples and the same initial test conditions and the samples stressed twice exhibited greater resistance to stress relaxation than those stressed only once. The phenomenon was found in samples that were unloaded for several weeks between tests, as well as in samples retorqued to their initial values while still loaded in compression. The effect of retorquing is shown in Fig. A-5 and A-6. At an initial stress level of 350 psi, the average rate of stress relaxation for a 24-hour period was 4.58 psi/hr. After retorquing, the same samples had an average rate of 0.062 psi/hr. The average rate of relaxation over a 100-hour period for samples initially stressed to 500 psi was 2.07 psi/hr. The same samples had an average value of 0.107 psi/hr after retorquing. Additional resistance to stress relaxation was found with each successive loading, but to a much lesser degree.

Further evidence confirming these data was obtained when load decay was measured for Kel-F liners installed in turbopump inlet housings. Load decay was measured for 24 hours; the bolts were then retorqued to their original value, and the load decay was again measured. Results indicated a much greater resistance to load decay after the retorquing.

Because of this supporting evidence, it was decided that the bolts loading the Kel-F would be retorqued to initial values after maintaining the Kel-F at elevated temperatures for a short duration.

Because preliminary tests indicated that a large portion of the stress relaxation of Kel-F occurred after 24 hours at 160 F, these conditions were used to accelerate the stress relaxation in Kel-F inlet liners during turbopump assembly. A further benefit, stress relief of the Kel-F which results in a more dimensionally stable part, is also obtained from this temperature cycle.


10,000
5,000
2,500
1,000
500
250
100
STRESS, PSI
0.1
0.5
1.0
5
10
TIME,
TEST CONDITIONS
ANNEALED KEL-F
TEST TEMPERATURE
160 F

1

R-5166

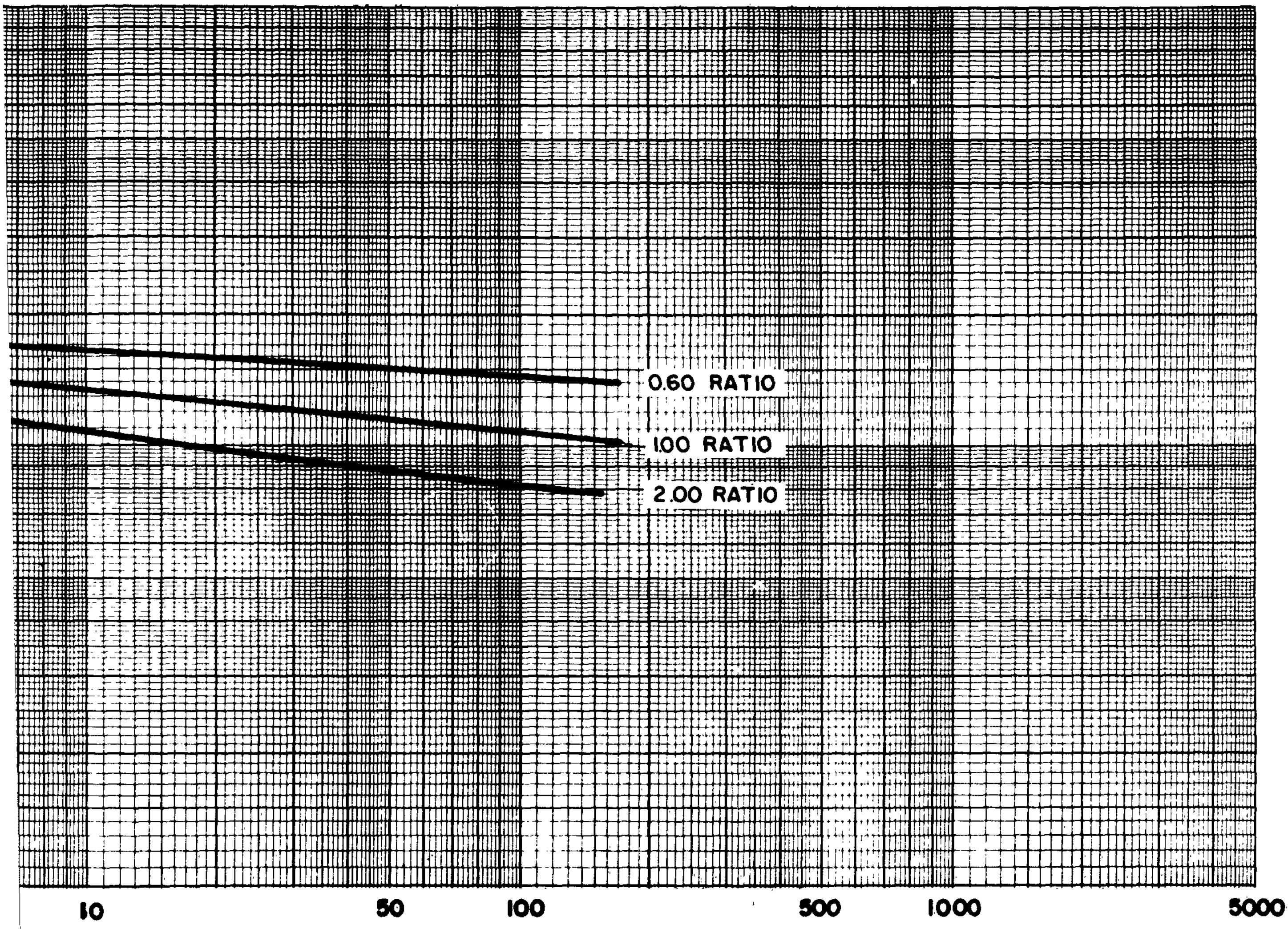
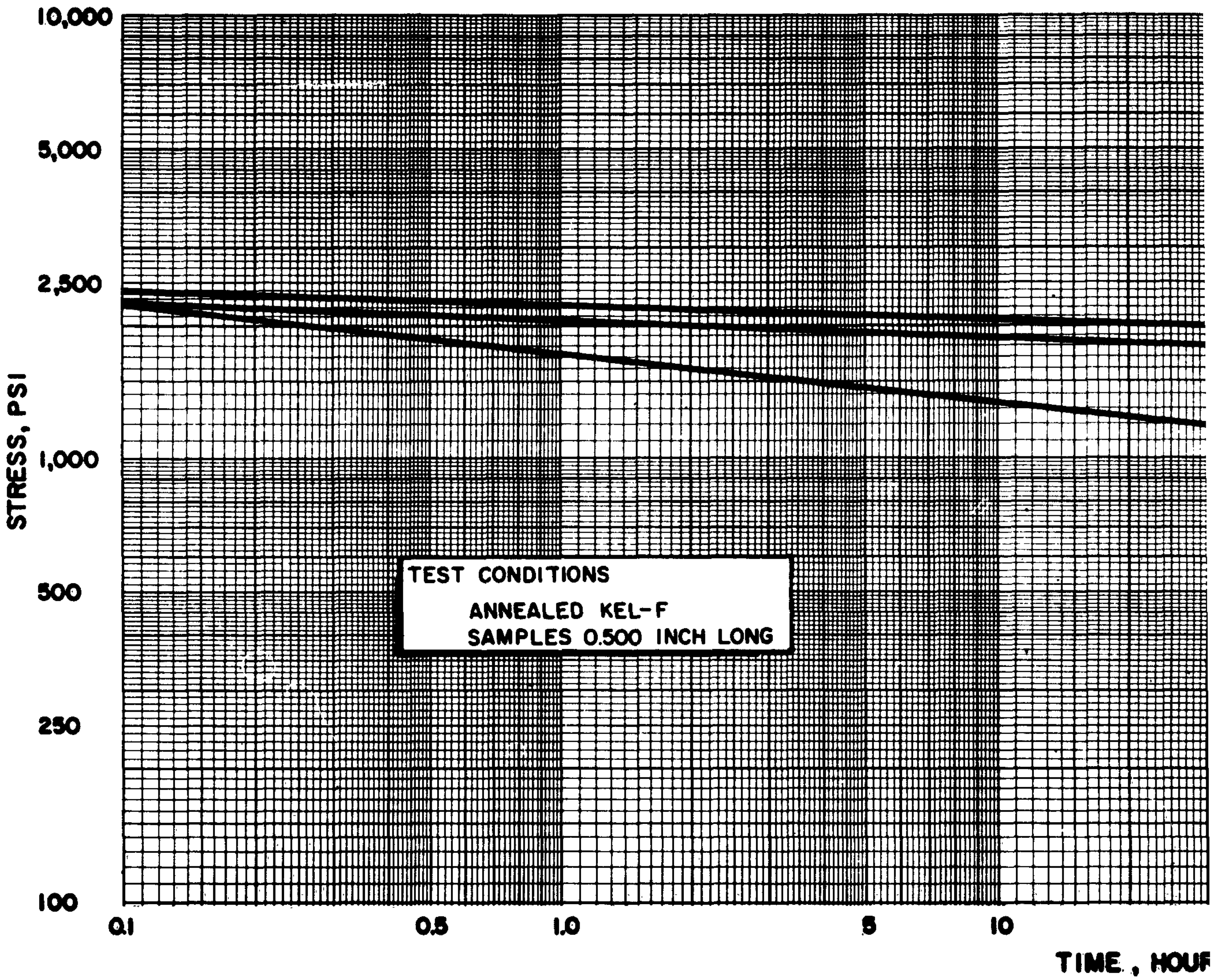


Figure A-2 . Effect of Length-to-Diameter Ratio on Stress Relaxation of Kel-F

2


10,000
5,000
2,500
1,000
500
250
100
STRESS, PSI
TEST CONDITIONS
ANNEALED KEL-F
SAMPLES 0.500 INCH LONG
0.1
0.5
1.0
5
10
TIME, HOUR

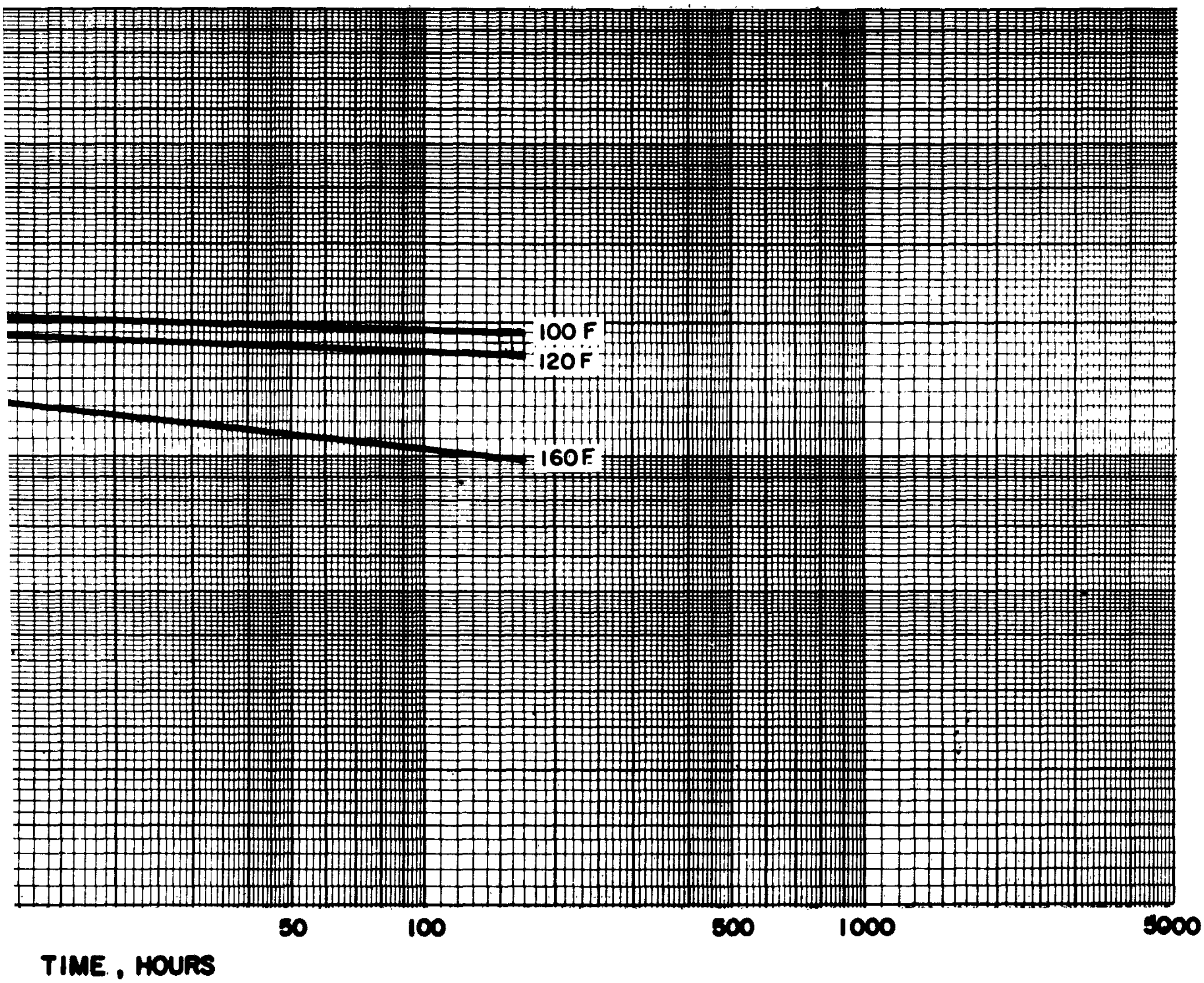
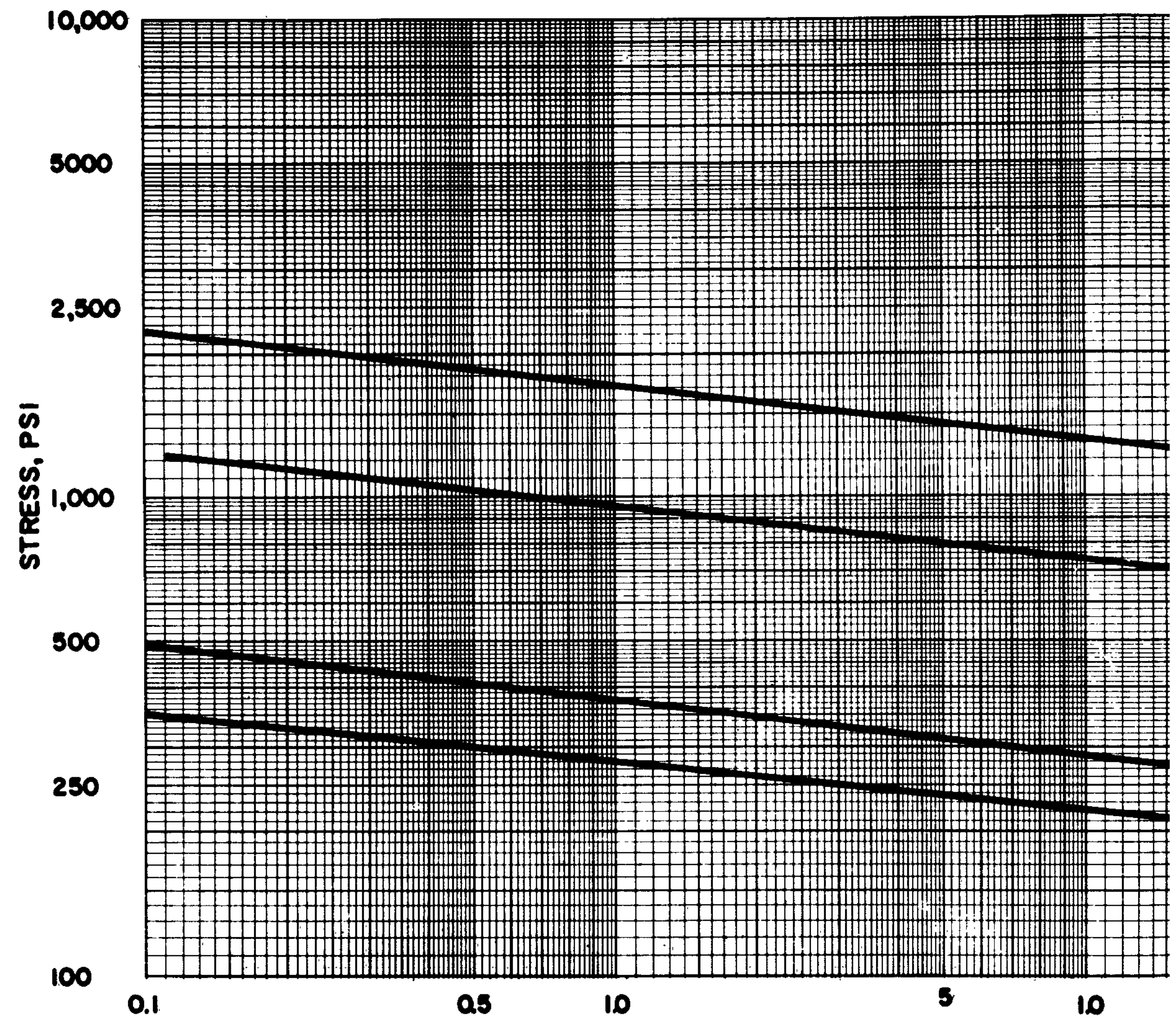


Figure A-3. Effects of Temperature on Stress Relaxation of Kel-F



1

R-5166

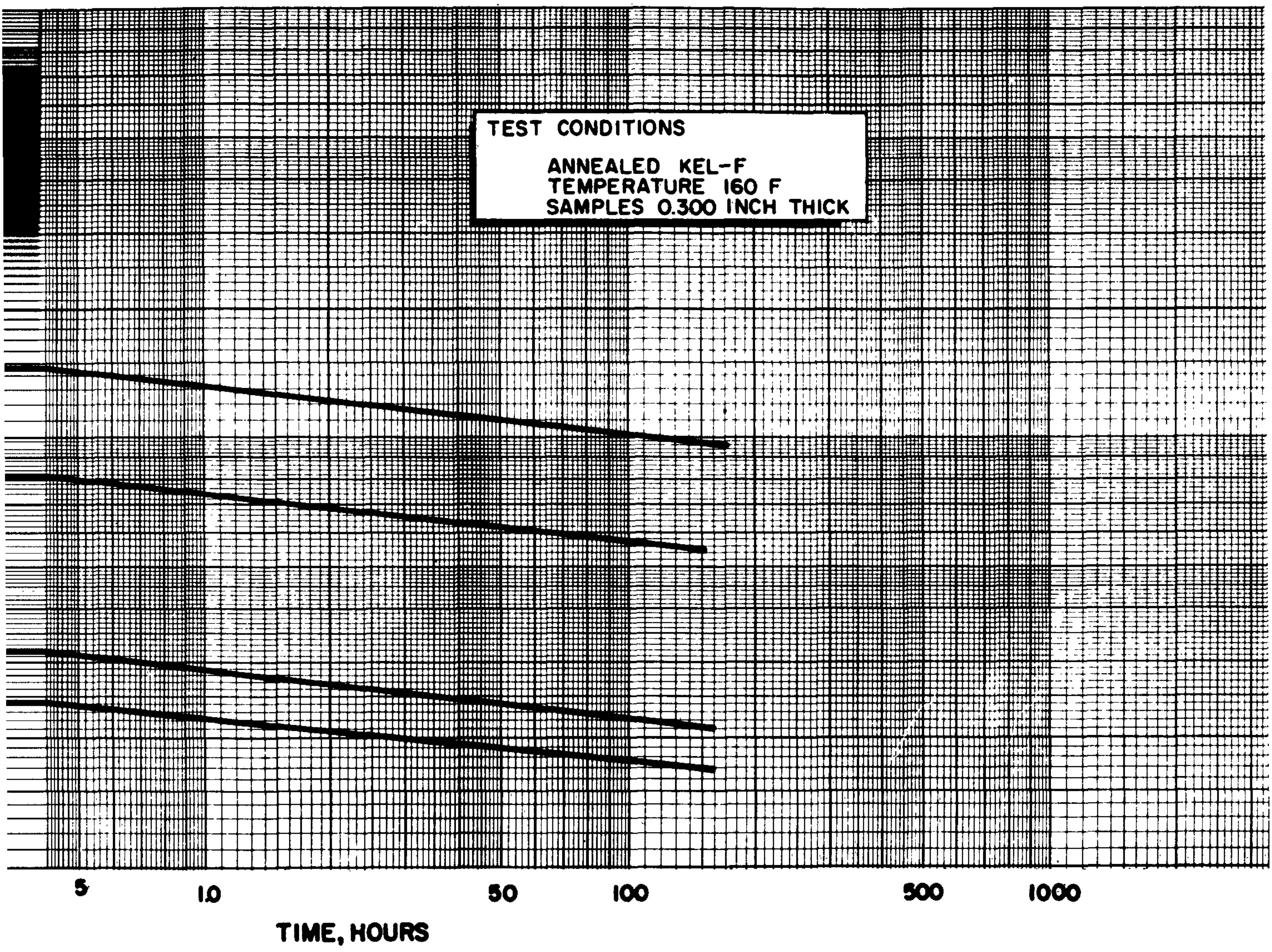
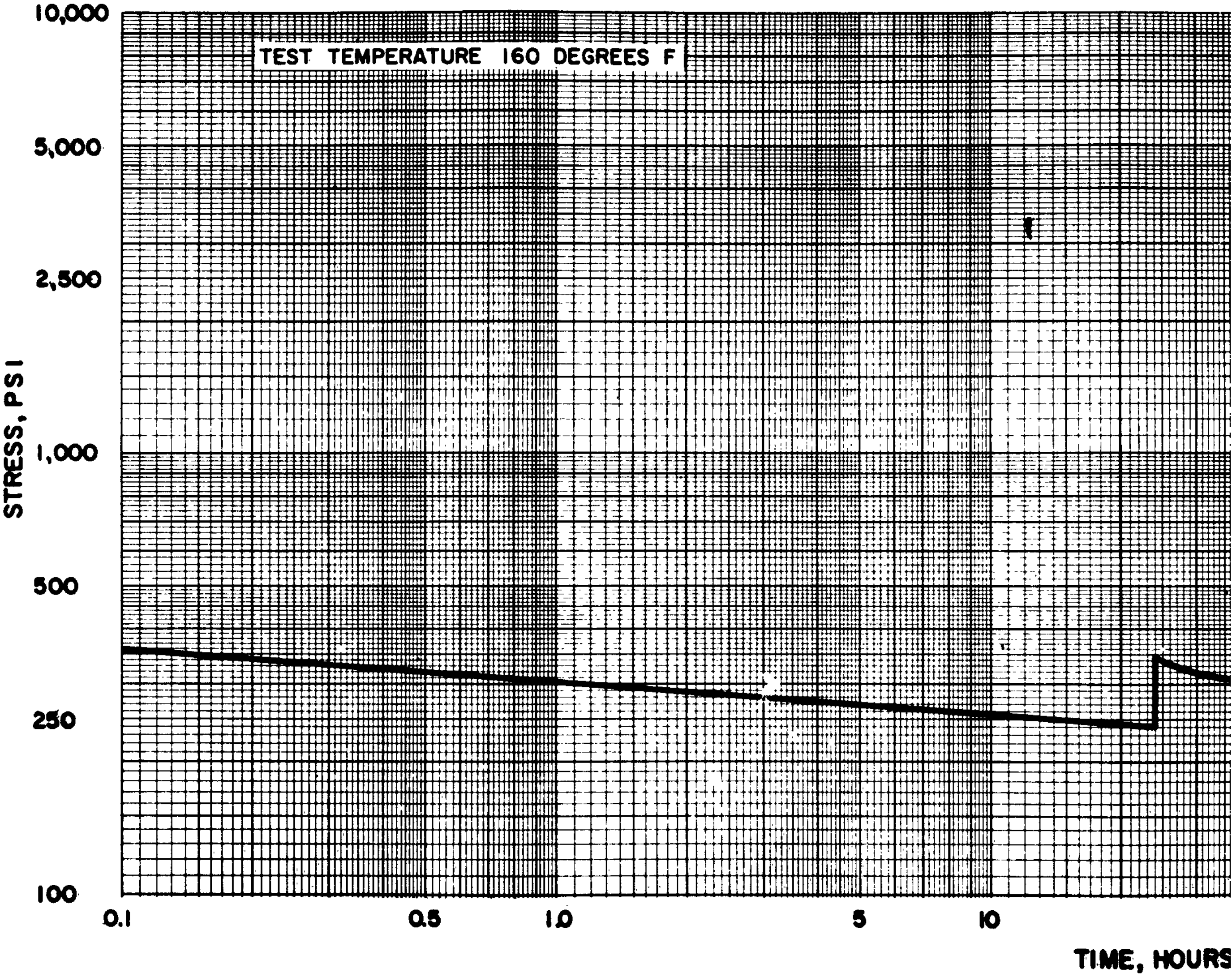
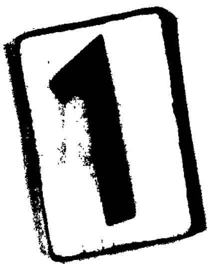


Figure A-4. Effects of Initial
on Stress Relaxati
Kel-F

2


TEST TEMPERATURE 160 DEGREES F
STRESS, PSI
10,000
5,000
2,500
1,000
500
250
100
0.1
0.5
1.0
5
10
TIME, HOURS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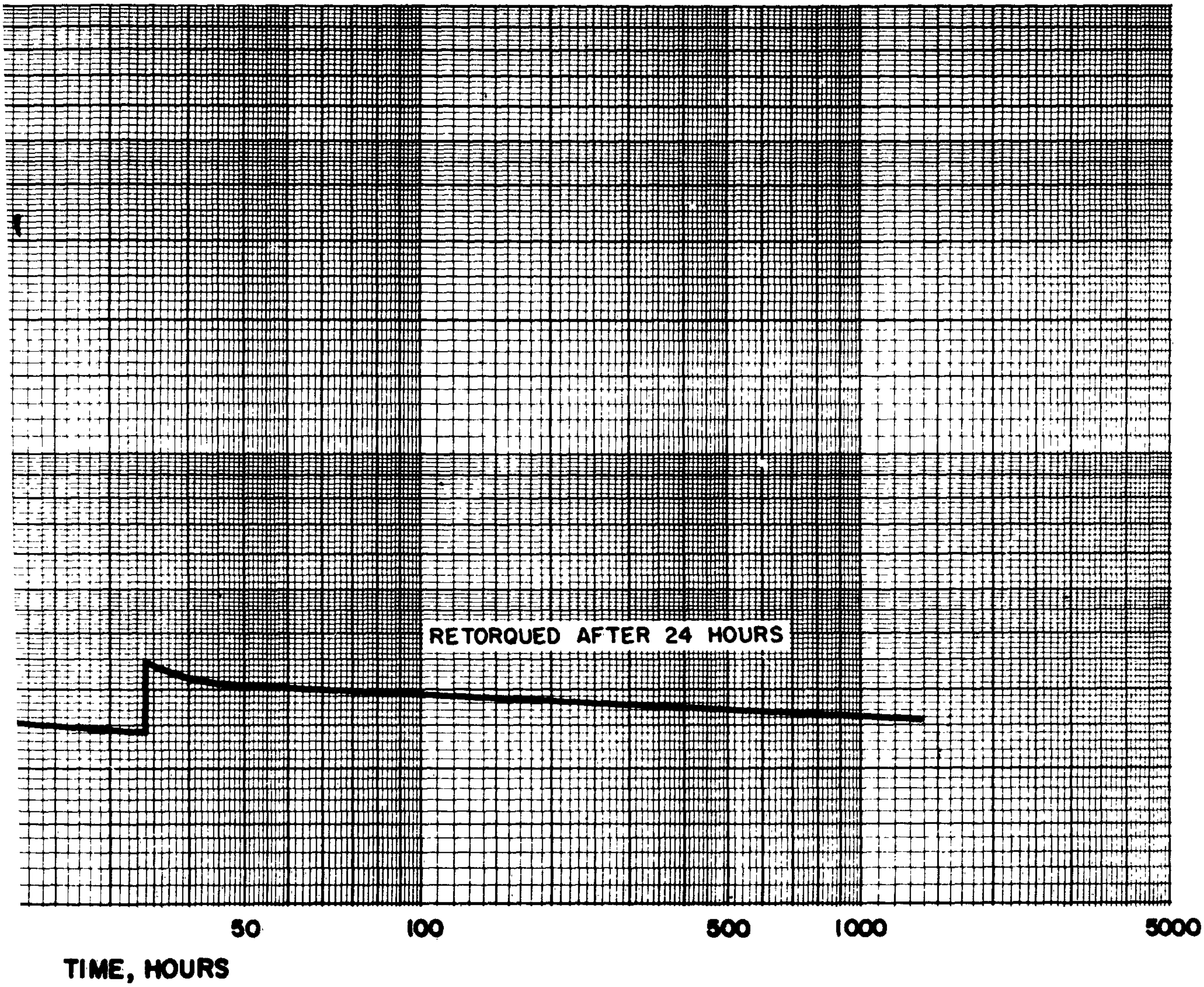
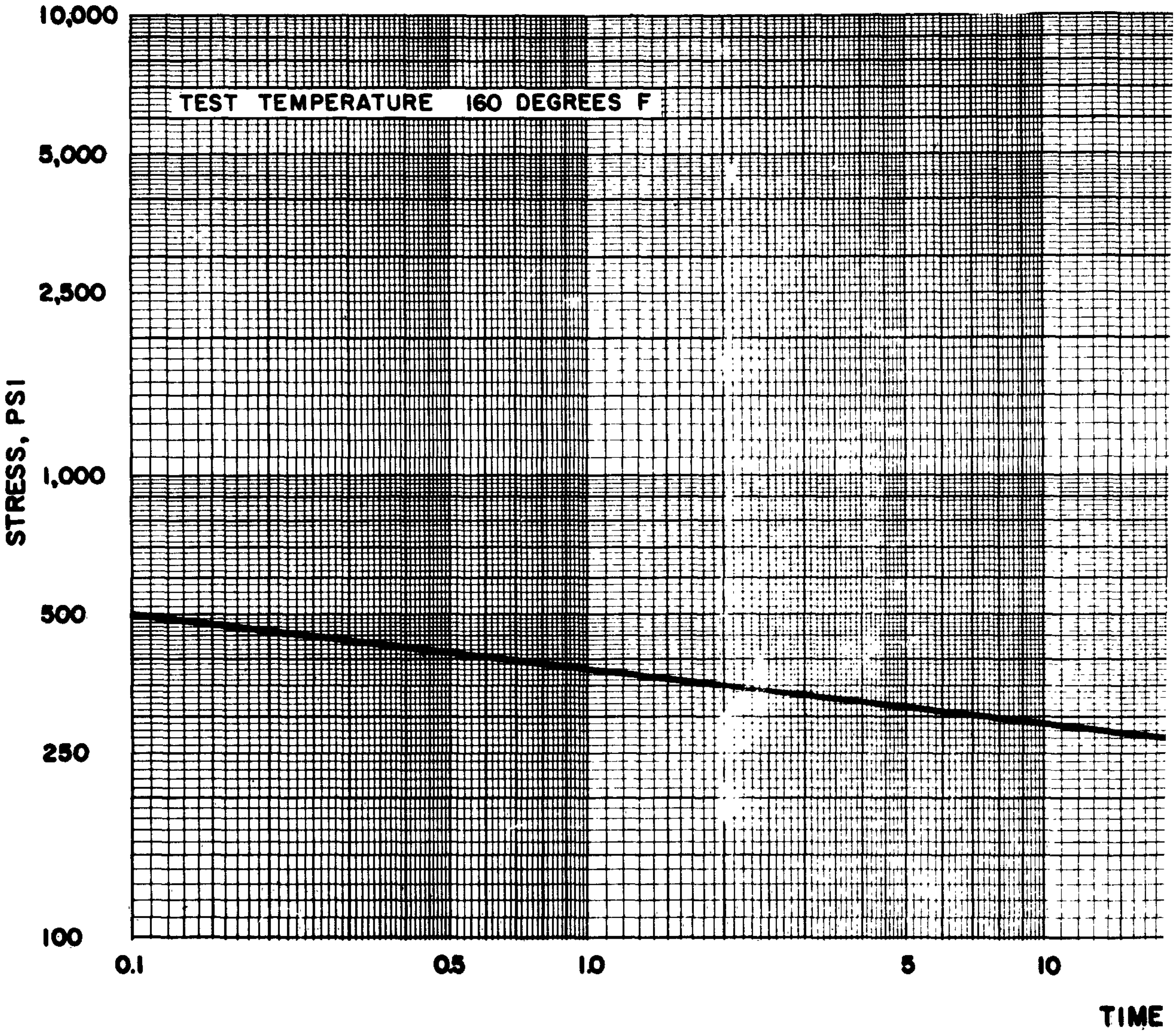


Figure A-5. Effects of Retorquing Kel-F After 24 Hours



TEST TEMPERATURE 160 DEGREES F
10,000
5,000
2,500
1,000
500
250
100
STRESS, PSI
0.1
0.5
1.0
5
10
TIME

1

R-5166

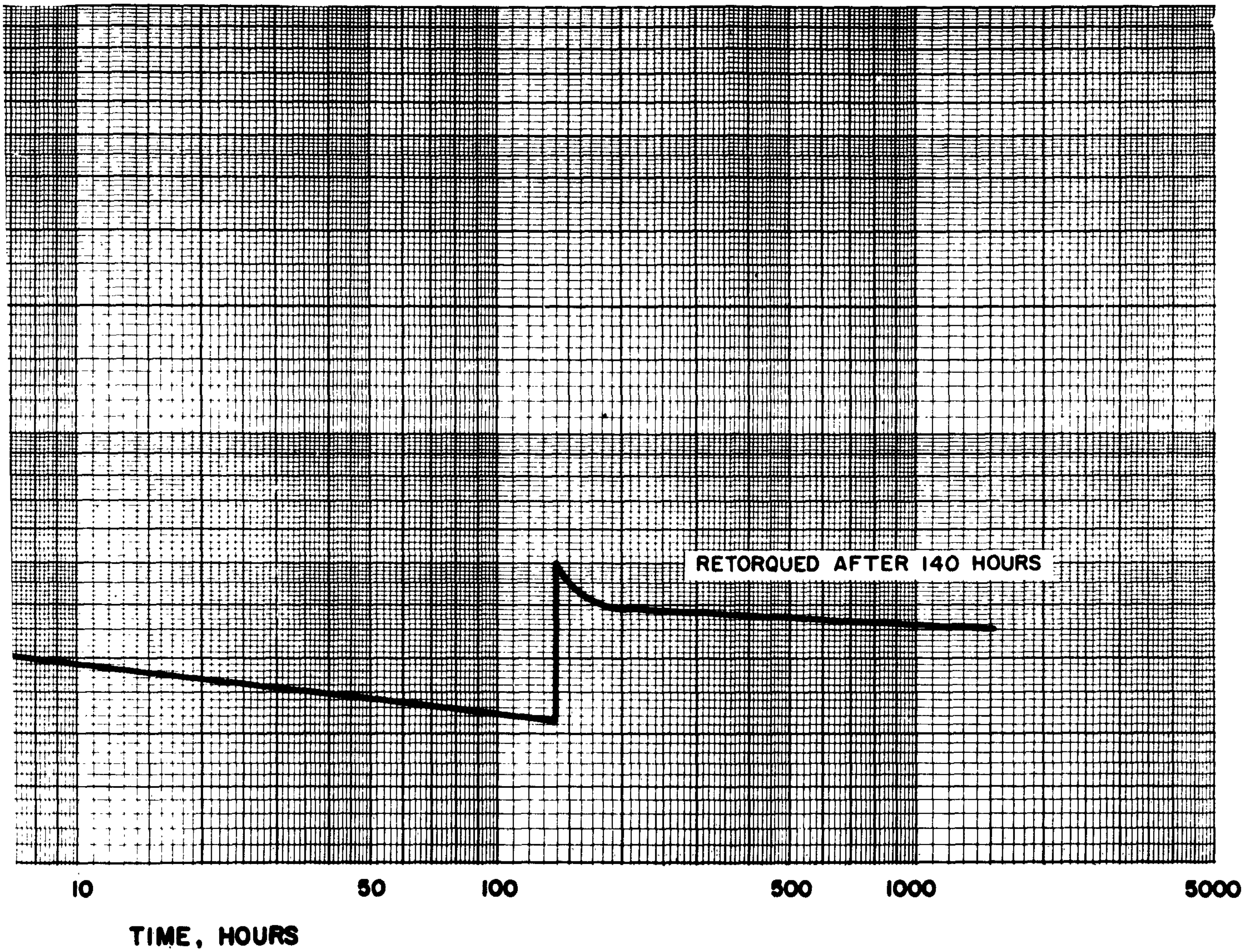


Figure A-6. Effects of Retorquing Kel-F After 140 Hours

APPENDIX B

DESIGN

PHILOSOPHY

A plastic-lined inlet was proposed for the Mark 4 LOX pump to eliminate the danger of explosion when inducer or impeller rubbing occurs. Although a Kel-F-lined inlet had previously been considered to lessen the dangers inherent with close-running metal-to-metal clearances, the amount of data concerning processing thick sections of Kel-F plastic was not sufficient to warrant confident use of the lined inlets.

During the development program, all applicable past experience and conditions of pump operation were reviewed. To perform satisfactorily, the inlet was required to fulfill the following conditions:

1. Perform reliably under all probable operating conditions associated with the turbopump, engine, and missile
2. Be capable of design and development within a short period of time
3. Demonstrate adequate storage and use life
4. Demonstrate hydrodynamic performance requiring no recalibration
5. Be capable of field retrofit
6. Produce no explosions as a result of impeller or inducer rubbing

DESIGN REQUIREMENTS

The final development design of the Kel-F-lined LOX pump inlet is defined by assembly drawing 458139-X3. The designation -X3 on the drawing represents changes affecting the diametral fits and clearances. All development testing described in this report was performed using the -X3 configuration.

The assembly consists of five basic parts; the Kel-F liner, the inlet housing, a support ring, the retaining bolts, and the bolt spacers. The individual parts are shown in Fig. B-1, and the assembled inlet is shown in Fig. B-2. Sixteen sets of bolts and spacers are required to secure the support ring and liner to the housing. The entire Kel-F lined LOX inlet assembly weighs 8.6 pounds, which is 1 pound less than the previous all-metal inlet assembly.

At completion of the development program, drawing 458385 was released for production use. Dimensions on the production drawing are identical with dimensions on the 458139-X3 development drawing, but the production drawings incorporate better definitions of manufacturing processes and procedures to ensure that the quality of parts obtained during development testing is maintained.

Kel-F Liner

The requirements to perform under all operating conditions necessitates that any plastic liner must (1) be compatible with LOX, (2) withstand pressure loading and fluid dynamic loads, (3) be incapable of causing explosions in the event of interference rubs, and (4) capable of some

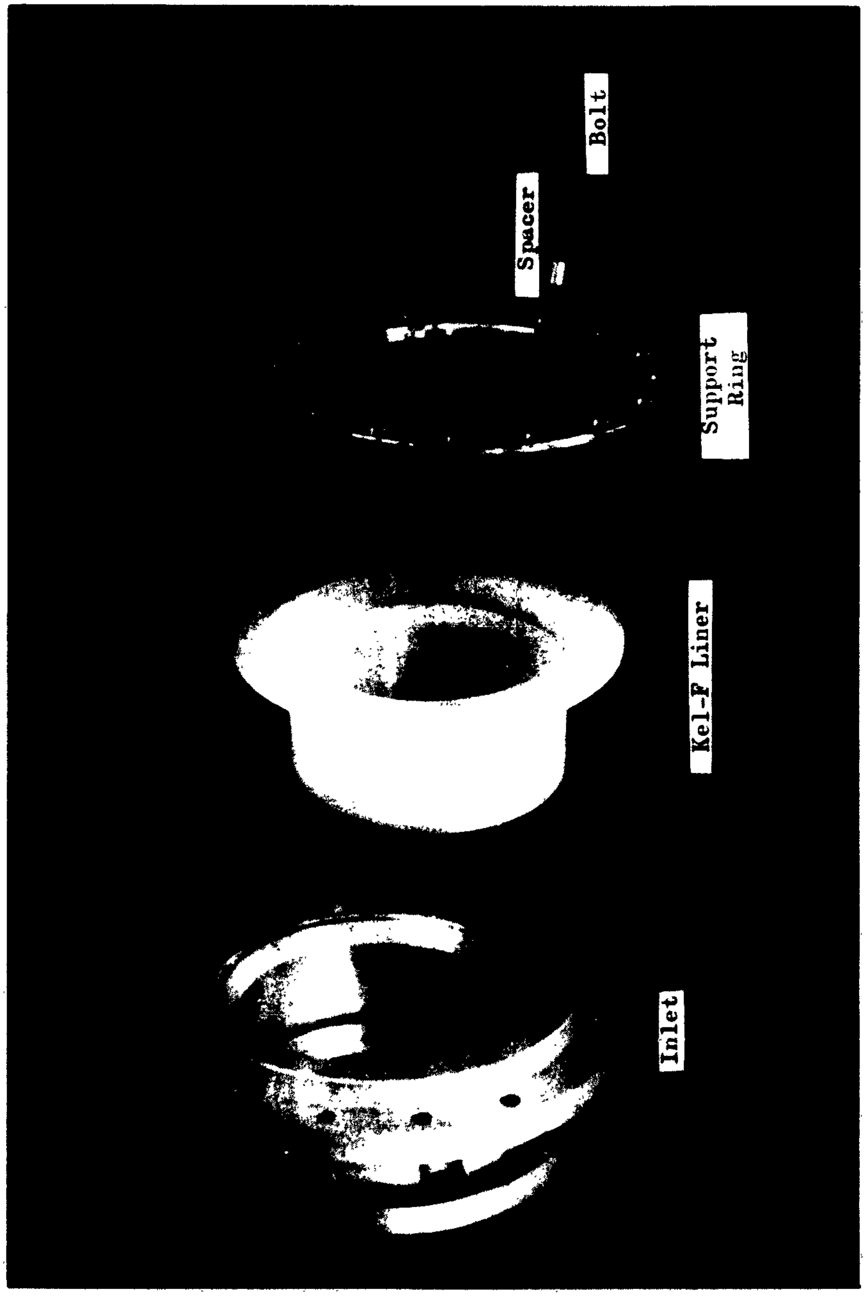


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Figure B-1. Mark 4 LOX Pump Inlet and Kel-F Liner Assembly

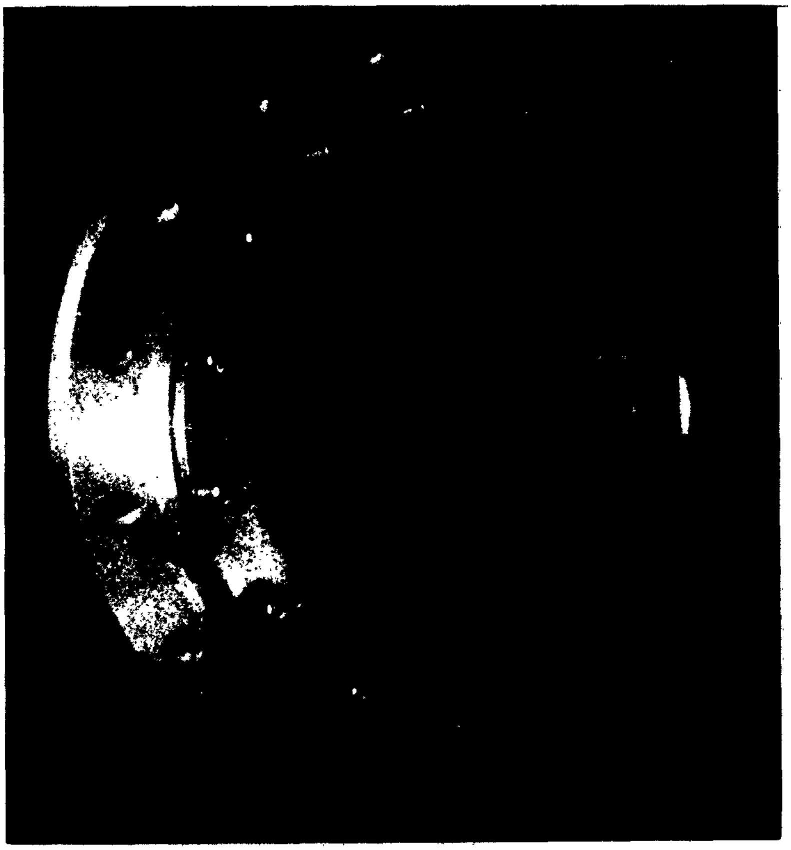

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Figure B-2. Kel-F-Lined Mark 4 LOX Pump Inlet resilience and plastic flow when subjected to interference by the impeller or inducer. To fulfill the primary condition (no detrimental effects when rubbed) the material Kel-F was chosen. This material has previously demonstrated LOX compatibility and is used extensively in the form of seals throughout LOX systems.

Kel-F is a thermoplastic resin formed by the homopolymerization of chlorotrifluoroethylene, and is symbolized by the chemical formula CF_2CFCl. Its chemical inertness is attributed to the high degree of fluorination. Kel-F demonstrates good mechanical properties at room temperature and exhibits a substantial improvement in strength at cryogenic temperatures. Although Kel-F is less ductile at cryogenic temperatures, the 70% crystallinity material exhibits approximately 1.5% elongation at -320 F. Some properties of Kel-F are shown in Table B-1.

Performance Considerations. To minimize performance changes, considerable effort was directed toward maintaining similar geometry between the Kel-F-lined inlet (Fig. B-3) and the original all-metal inlet (Fig. B-4). Two labyrinth steps were removed to reduce the unsupported span of Kel-F exposed to the differential pressure resulting from the pump discharge pressure and the reduced pressure through the labyrinth, and the radial assembly clearance was reduced by 0.002 inch to prevent a reduction in performance. The reduction in radial clearance was permissible because rubbing between the Kel-F liner and impeller produces no detrimental effects. In addition, the diverter lip on the all-metal inlet was deleted from the Kel-F liner to simplify the design and preclude the possibility that the thin lip might crack off when subjected to rubbing conditions at cryogenic temperatures.

TABLE B-1

PROPERTIES OF KEL-F AT VARIOUS TEMPERATURES

Temperature, F	Properties at 70% Crystallinity*					
	Ultimate Tensile Strength, psi	Elongation, %	Modulus of Elasticity in Tension, psi	Compressive Strength, psi	Modulus of Elasticity in Compression, psi	Tensile Yield, psi
-320	15,600	1.5	760,000	38,000	1,650,000	15,000
-100	12,000	10.0	500,000	25,000	690,000	12,000
+77	4,600	46.0	220,000	8,500	330,000	6,500

*Crystallinity is a measure of crystallites formed when a polymer is cooled below its melting point. Rapid quenching produces a low percentage of crystallinity.

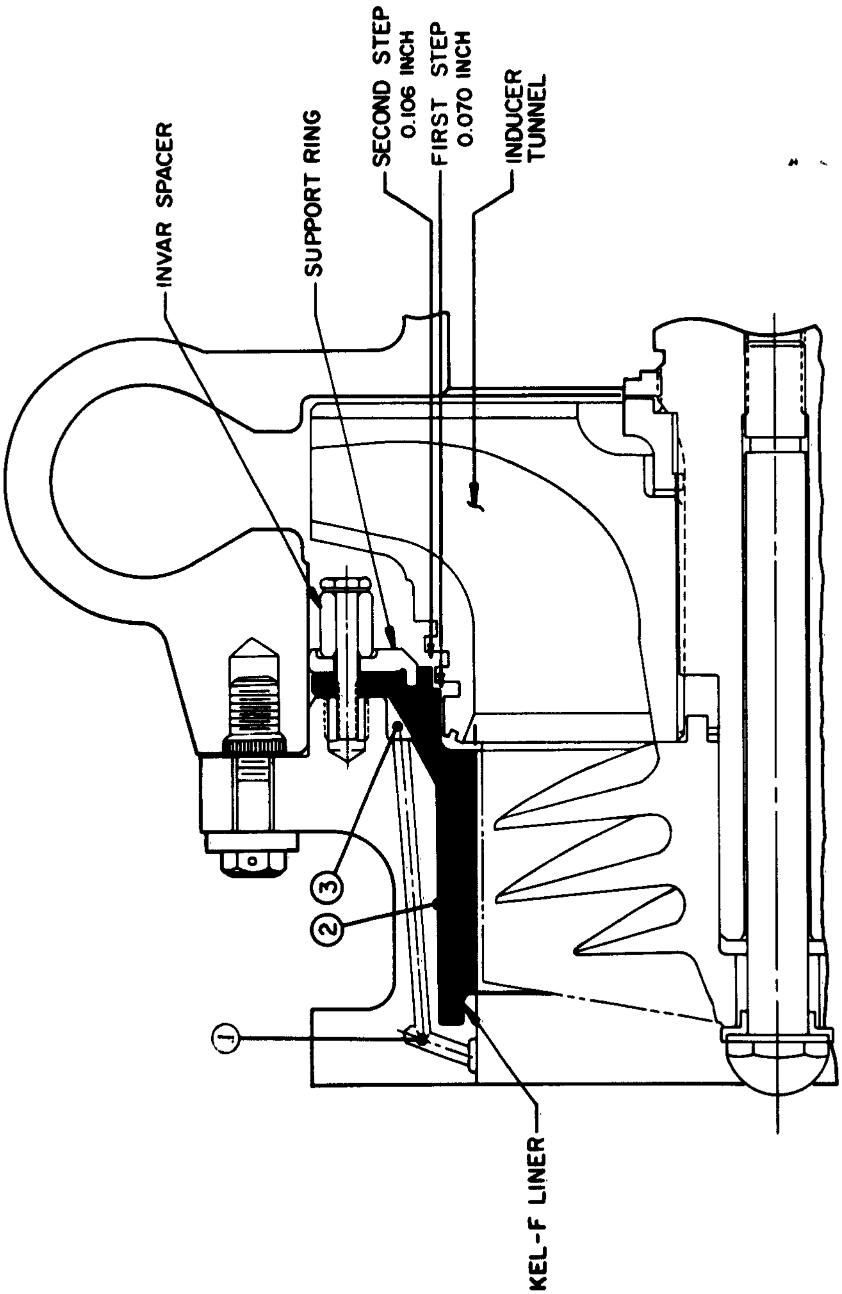

Figure B-3. LOX Pump and Kel-F-Lined Inlet Adapter

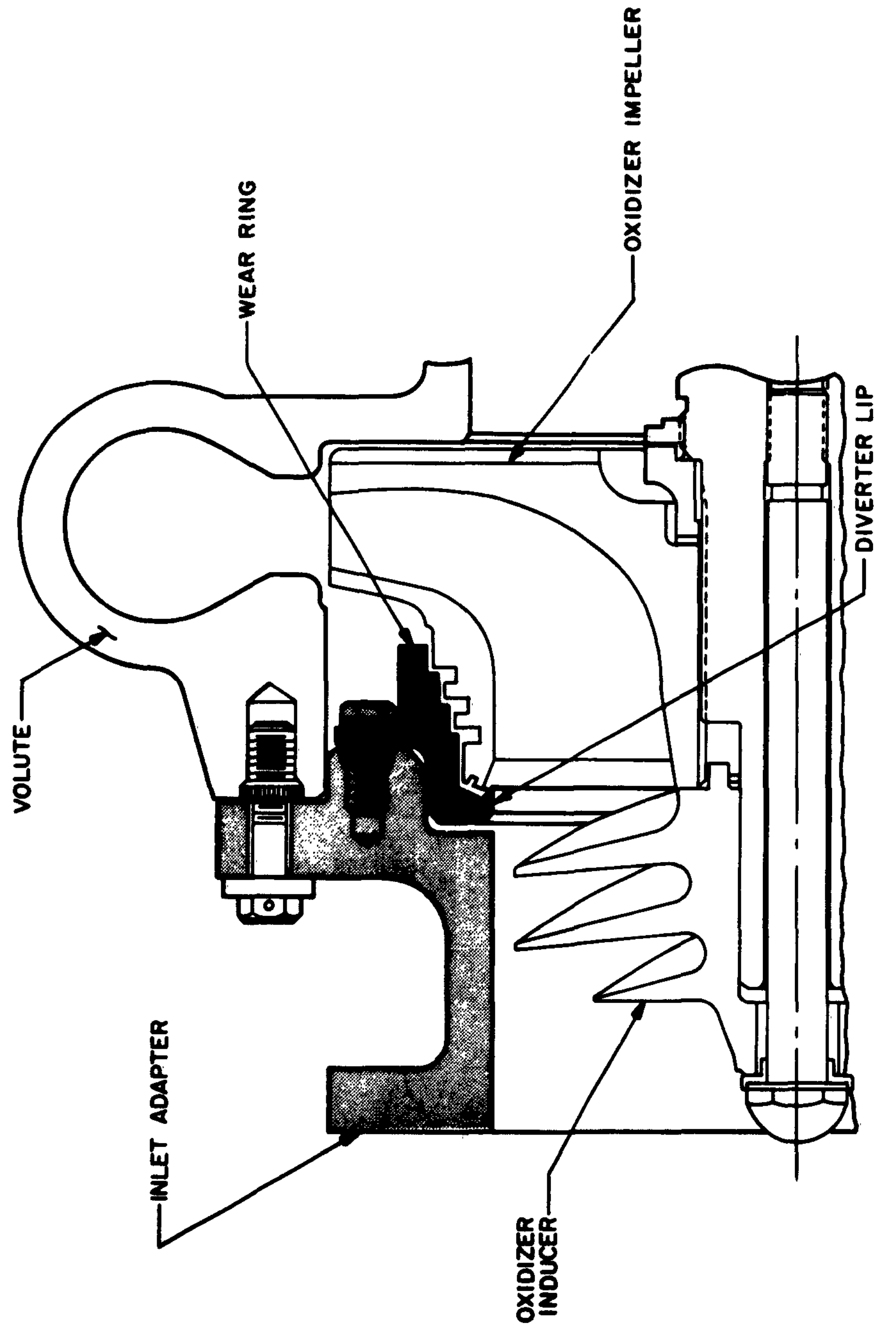

Figure B-4. Original Configuration of LOX Pump and Inlet Adapter

Inlet Housing

The housing surrounding the Kel-F liner was machined from a forging blank of type 6160 aluminum. This material was selected to match the properties of the aluminum pump volute to which the housing attaches. The use of aluminum also minimizes weight and allows thermal contraction characteristics compatible with those of the Kel-F liner.

To protect against the possible buildup of hydrostatic pressure between the liner and housing, eight 1/8-inch-diameter passages were installed within the housing. These passages vent the collector annulus located near the high-pressure end of the liner. The passages are visible in Fig. B-3.

Support Ring

A metal support ring with a centering pilot is used to clamp the liner to the housing. This ring prevents deformation of the liner caused by the differential pressure in the labyrinth area, and maintains concentricities of the plastic liner. The ring is machined from type 321 corrosion-resistant steel plate.

The outside diameter of the ring is a slip fit within the pump volute to prevent the metal support ring from shifting positions. The clamping action of the bolted-down support ring secures a Kel-F seal between the high-pressure pump discharge and the collector annulus. This Kel-F seal is not required to be leak tight, but must maintain the leakage at a level which can be adequately drained by the vent passages to prevent excessive pressure buildup. This design function of the vent passages was substantiated during the test program.

Retaining Bolts and Spacers

A total of 16 bolts are used to secure the support ring and liner to the housing. This number of bolts was chosen to ensure an even distribution of load over the face of liner flange and to permit minimum thickness of the retainer ring. Even loading over the liner flange prevents concentrated stresses within the plastic and also ensures a better Kel-F seal to prevent excessive leakage.

The retaining ring bolts are made of type A286 steel and are equipped with Invar spacers to minimize the differential expansion problems associated with the different thermal expansion properties of the materials used. The length of the Invar spacer is accurately maintained so that a relatively constant preload stress will be obtained within the bolt over the temperature range of +160 to -320 F.

Kel-F has a comparatively large change in size for a given temperature variation, and the Invar spacer is required so that the bolt length can be increased to allow changes comparable to changes in Kel-F over the operating temperature range of the inlet assembly. Invar was chosen for the spacer because it exhibits a small thermal expansion characteristic which will not nullify the contraction or expansion of the bolt. The bolts holding the retaining ring and liner are prevented from rotating by using locking inserts installed within the housing.

APPENDIX C

MARK 4 TURBOPUMP CRITICAL SPEED STUDY

The purpose of this study was to determine how various dynamic considerations will affect critical speeds of the Mark 4 turbopump shaft. These dynamic effects, which were assumed of negligible importance in previous studies, included:

1. Rotary inertia of the LOX impeller-inducer mass
2. Gyroscopic action of the LOX impeller-inducer mass
3. Various ball bearing spring rates, particularly nonlinear spring rates
4. Static loading conditions (i.e., the gear load and the LOX impeller load)

For the study, the shaft (Fig. C-1) was considered as the lumped-mass system shown in Fig. C-2. The characteristics of the lumped parameters were as follows:

I_1 = mass moment of inertia (I_{zz}) of all rotating components extending from the LOX end of the shaft to the slinger

M_2 = mass of the components of I_1 concentrated at the center of gravity of the components

M_3 = mass of all rotating components extending 0.75 inch from the fuel impeller exit to 1.50 inches from the fuel impeller entry face. The mass was considered as a point mass located at the center of gravity of the components

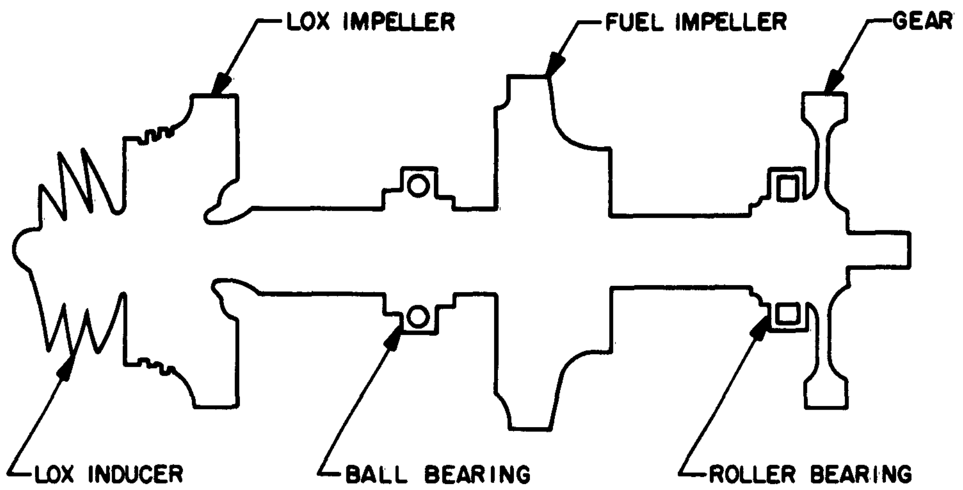

Figure C-1. Mark 4 Rotor

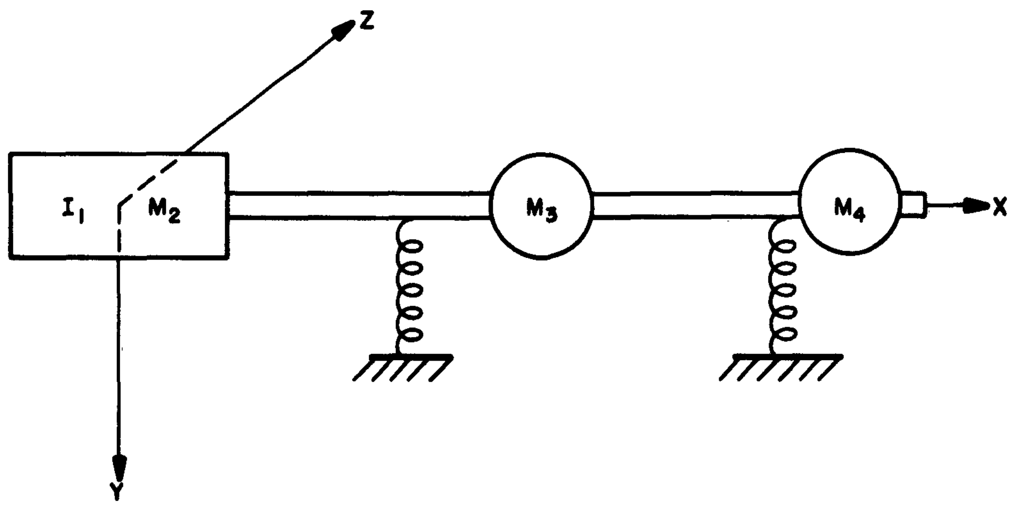

Figure C-2. Mark 4 Mathematical Model

M_4 = mass of all rotating components extending from the roller bearing to the fuel end of the rotor. This mass was also considered as a point mass located at the center of gravity of the components.

The equations of motion of the lumped-mass system were developed assuming that the masses were connected by weightless, flexible links. The flexibility of the links reflected the actual flexibility of the shaft. It was also assumed that the shaft was symmetric.

Two different nonlinear ball bearing spring rates were studied and are shown in Fig. C-3 and C-4 . The two linear spring rates indicated in Fig. C-3 were also studied. It was originally intended to study the effects of a ball bearing spring rate which had a perfect dead zone as indicated by the dashed line in Fig. C-4, but because of computer noise, a very soft spring rate was added to keep the rotor in the center position for a no-load, equilibrium condition. The roller bearing spring rate was assumed linear and equal to a value of 3.97×10^6 lb/in. for all computer work.

During the time that the equations of motion of the shaft were being mechanized on the analog computer, preliminary calculations for shaft critical speeds were carried out to determine the relative effects of gyroscopic coupling and rotary inertia and to provide checks for the computer work.

The first natural frequency calculations for the shaft neglected the gyroscopic coupling and rotary inertia effects and assumed rigid bearings. The

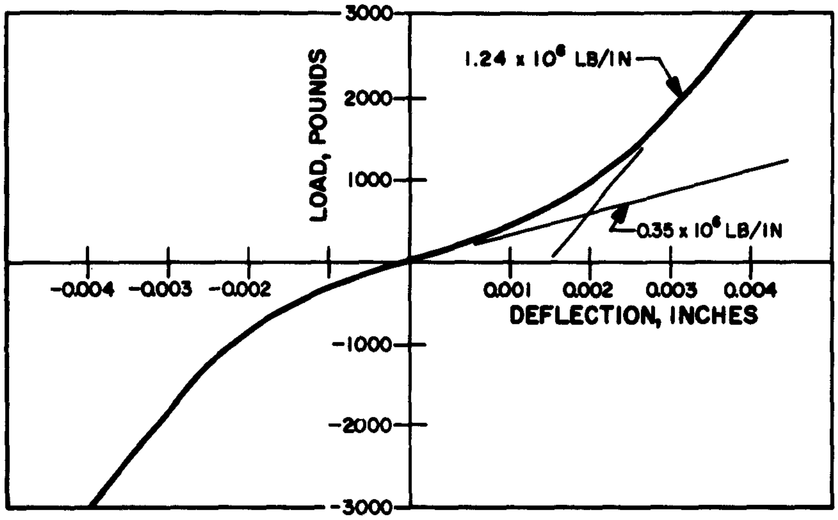

Figure C-3. Ball Bearing Spring Rate
Soft Nonlinearity

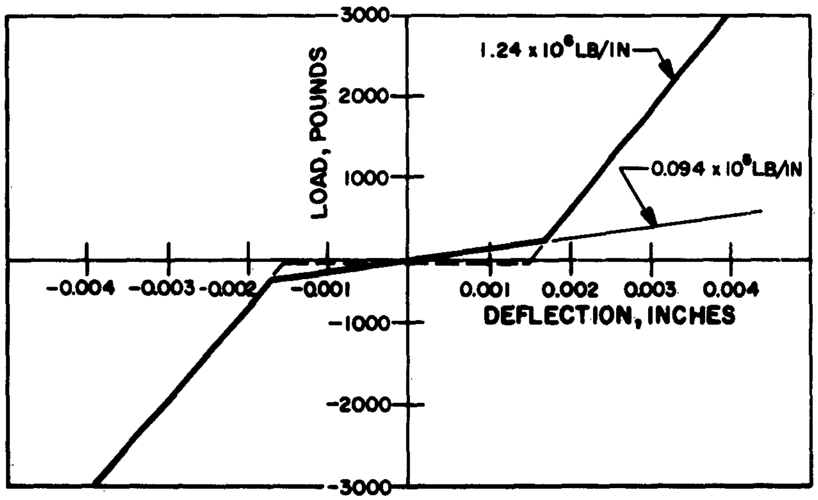

Figure C-4. Ball Bearing Spring Rate
Hard Nonlinearity

fundamental critical shaft speed for this case was 21,200 rpm. Adding the rotary inertial effect of the overhung mass lowered this critical speed (by about 8%) to 19,600 rpm.

The flexibility of the bearings was also considered in the preliminary work. Assuming a ball bearing spring rate of 0.35×10^6 lb/in. (corresponding to the lowest spring rate shown in Fig. C-3) and a roller bearing spring rate of 3.97×10^6 lb/in. gave a fundamental critical speed of 15,200 rpm. This critical speed reflects the rotary inertia effect, but does not include the gyroscopic coupling effects. Changing the spring rate of the ball bearings to 1.24×10^6 lb/in. (corresponding to the highest spring rate shown in Fig. C-3) changed the fundamental critical speed to 17,940 rpm.

The effects of three different ball bearing spring rates were investigated extensively on the analog computer. These spring rates were:

1. The linear spring rate of 1.24×10^6 lb/in.
2. The soft nonlinearity of Fig. C-3
3. The hard nonlinearity of Fig. C-4

The roller bearing spring rate of 3.97×10^6 lb/in. was used for all computer work.

To determine the relative accuracy of the computer mechanization, the first three critical speeds of the shaft for the linear ball bearing spring rate of 1.24×10^6 lb/in. were compared with the corresponding calculated critical speeds. The comparison is presented below (gyroscopic coupling effect was neglected).

Mode	Calculated Critical Speeds, rpm	Analog Results Critical Speeds, rpm
1	17,940	18,000
2	61,860	63,000
3	96,540	96,000

This comparison shows that the computer mechanization was correct. Including the gyroscopic coupling effect with the mechanization of the above problem raised the fundamental critical speed from 18,000 to 18,650 rpm. This demonstrates the stiffening effect of the gyroscopic action.

To reduce transient effects and to obtain finite motion at resonance, viscous damping was incorporated with the equations of motion mechanized on the computer. The over-all value used for damping was 1.5% of critical for the first mode of shaft with linear ball bearing spring rates. With nonlinear spring rates, the damping was amplitude-dependent and the highest damping rates occurred with low amplitudes. The maximum values of damping observed for the nonlinear cases were 1.9% first-mode damping for the soft nonlinearity, and 4.4% first-mode damping for the hard nonlinearity. These values of damping reflect the same mechanisms and dashpot values used for the linear case.

The fundamental critical speeds for the shaft with the nonlinear ball bearing spring rates were obtained utilizing a 0.3 in.-oz unbalance of the LOX impeller-inducer combination. This unbalance was approximately equivalent to a 0.002-inch offset of the principal axis of the mass.

The fundamental critical speed of the shaft obtained by including the soft nonlinear ball bearing spring rate, but excluding the static loading effects, was 17,750 rpm. Addition of the nominal static loading conditions raised the critical speed slightly (to 17,800 rpm). This increase reflects the preset in one plane of the bearings caused by the static loads. In both instances, a mild resonant condition was encountered at approximately 15,000 rpm. This minor resonance was a reverse whirl mode that was quite unstable inasmuch as the reverse whirl motion would not persist. Shaft deflections at this speed remained minor.

The hard nonlinear ball bearing spring rate had an interesting effect on the critical speeds of the shaft. Without the static loads, the critical speed of the shaft with the hard nonlinearity was 28,800 rpm. At this shaft speed a subharmonic resonance occurred, indicating that the shaft fundamental frequency was equivalent to 15,300 rpm. However, only a very minor resonant condition near a shaft speed of 15,300 rpm could be excited by the unbalance forces. Incorporation of the nominal static loads raised the critical speed to 31,500 rpm where the subharmonic response frequency was equivalent to 15,600 rpm.

A summary of critical speeds determined during this study is presented in Table C-1. Figures C-5 and C-6 show the maximum deflections of the shaft as a function of shaft speed. The variations in these deflections were caused by different values of unbalances, and are shown in Fig. C-7 through C-12.

FORM 608-B PLATE REV. 1-58

TABLE C-1

CRITICAL SPEEDS FOR THE MARK 4 TURBOPUMP SHAFT-ANALOG STUDY RESULTS

Ball Bearing Stiffness, lb/in.	Roller Bearing Stiffness, lb/in.	Overhung Inertia Effect	Gyroscopic Effect	Nominal Static Loads	Fundamental Critical Speed, rpm
Rigid	Rigid	Excluded	Excluded	No Effect	21,200
Rigid	Rigid	Included	Excluded	No Effect	19,600
0.35×10^6	3.97×10^6	Included	Excluded	No Effect	15,200
1.24×10^6	3.97×10^6	Included	Excluded	No Effect	18,000
1.24×10^6	3.97×10^6	Included	Included	No Effect	18,650
Nonlinear (soft)	3.97×10^6	Included	Included	Excluded	17,750[1]
Nonlinear (soft)	3.97×10^6	Included	Included	Included	17,800[1]
Nonlinear (hard)	3.97×10^6	Included	Included	Excluded	28,800[1,2]
Nonlinear (hard)	3.97×10^6	Included	Included	Included	31,500[1,3]

[1]These critical speeds were obtained for 0.3 in.-oz unbalance of the LOX impeller

[2]Subharmonic resonance-response frequency was 255 cps (15,300 rpm)

[3]Subharmonic resonance-response frequency was 260 cps (15,600 rpm)

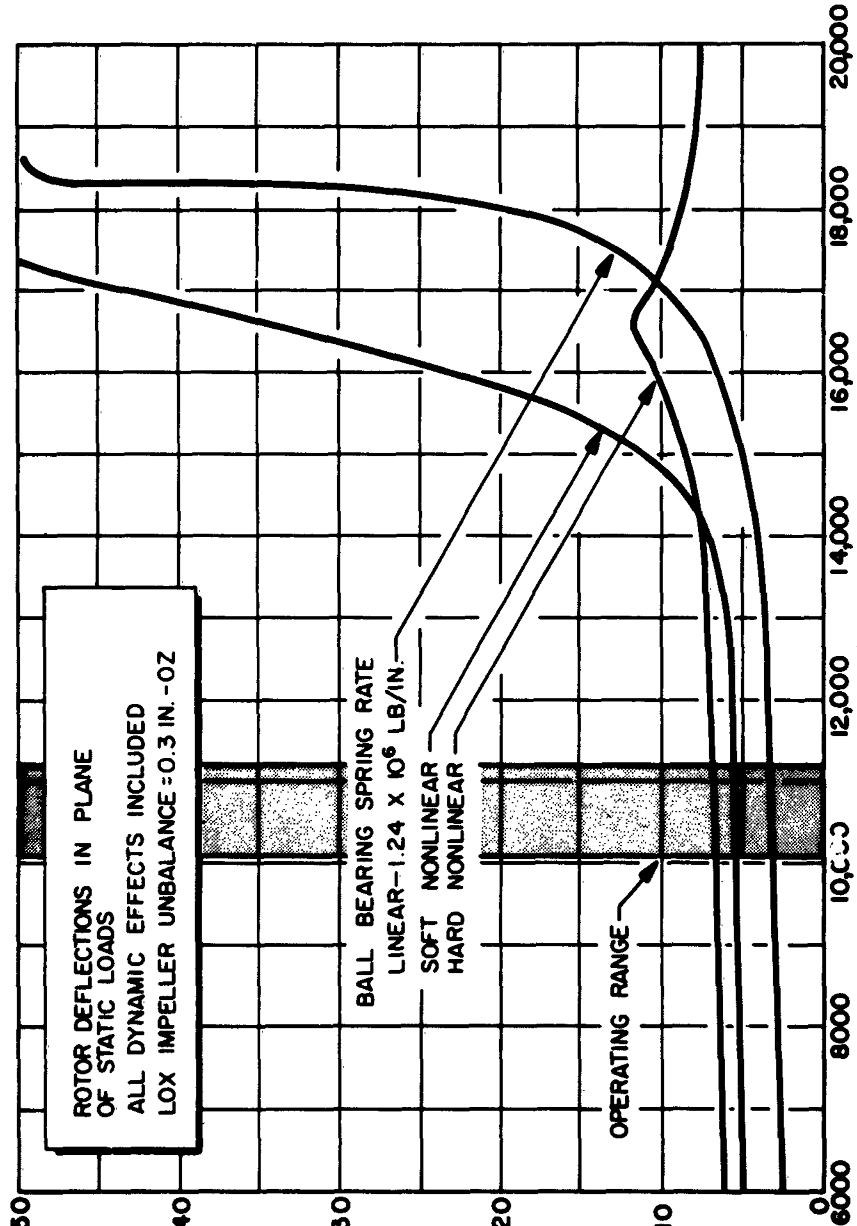

Figure C-5. Rotor Deflections in Plane of Static Loads, LOX Impeller Unbalance = 0.3 in.-oz

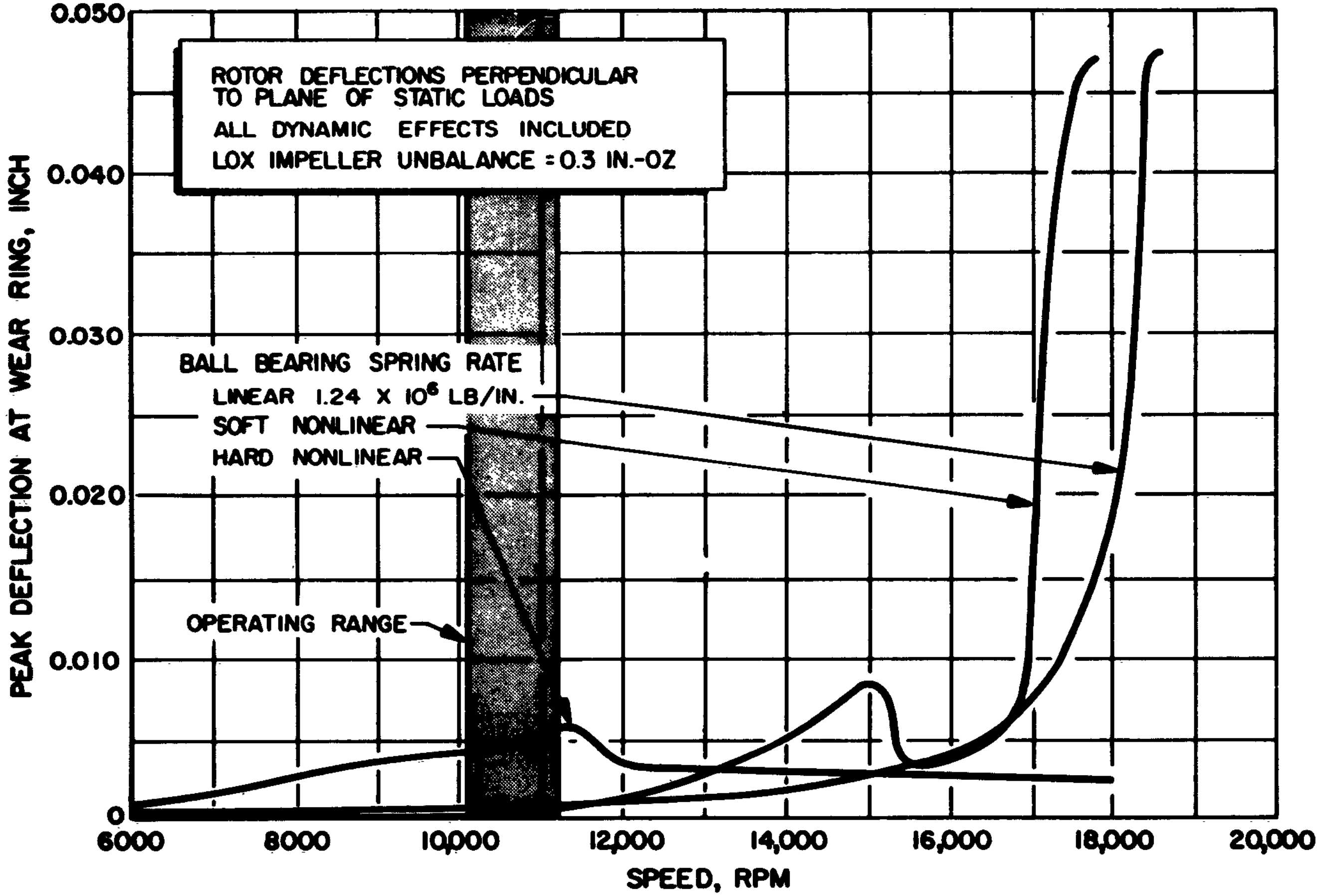

Figure C-6. Rotor Deflections Perpendicular to Plane of Static Loads, LOX Impeller Unbalance = 0.3 in.-oz

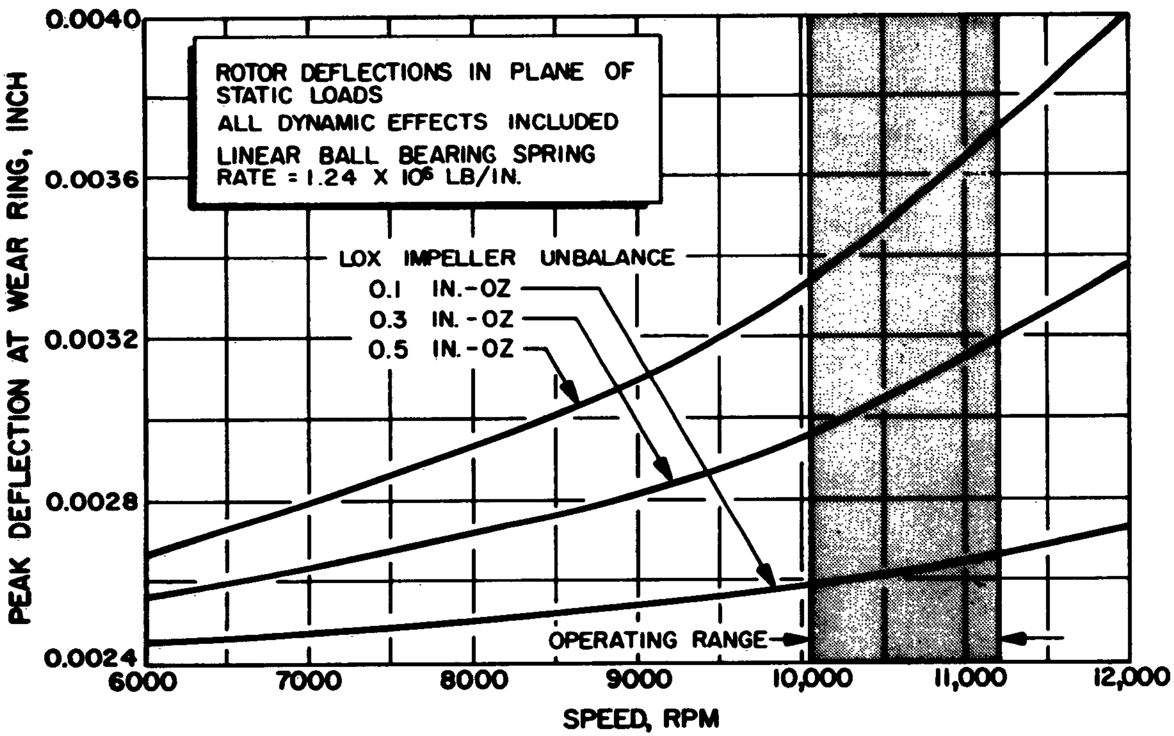

Figure C-7. Rotor Deflections in Plane of Static Loads, Linear Ball Bearing Spring Rate = 1.24×10^6 lb/in.

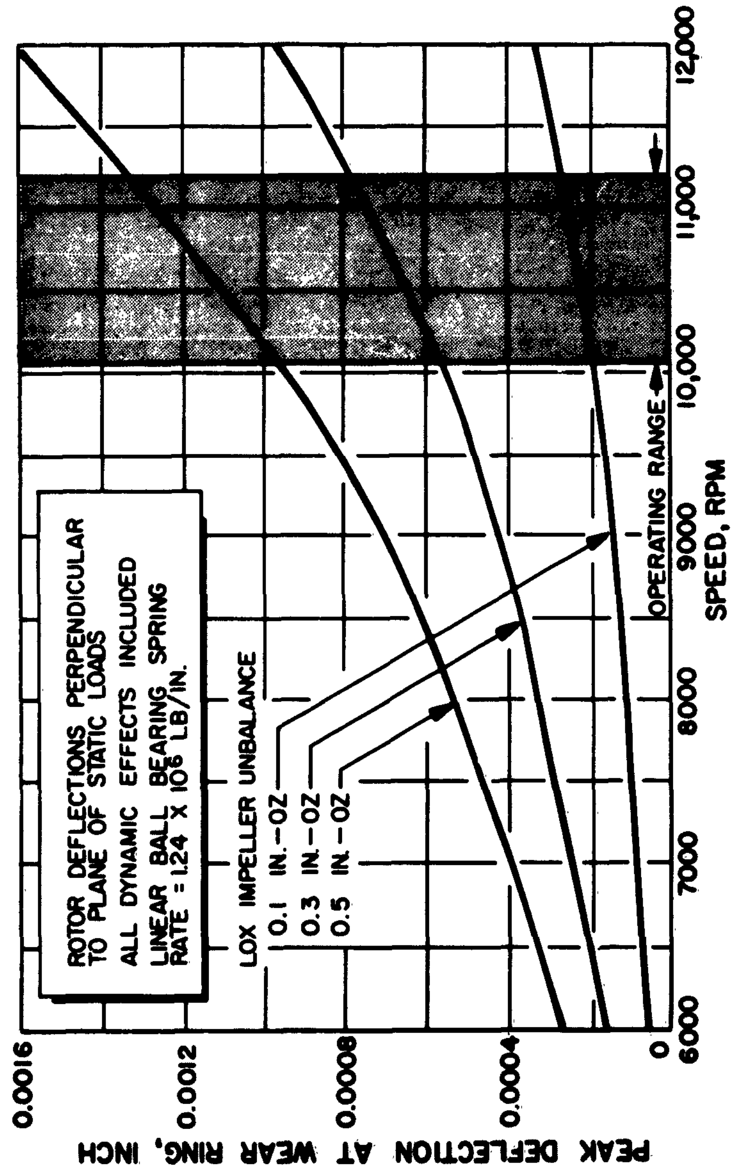

Figure C-8. Rotor Deflections Perpendicular to Plane of Static Loads, Linear Ball Bearing Spring Rate = 1.24 x 10^6 lb/in.

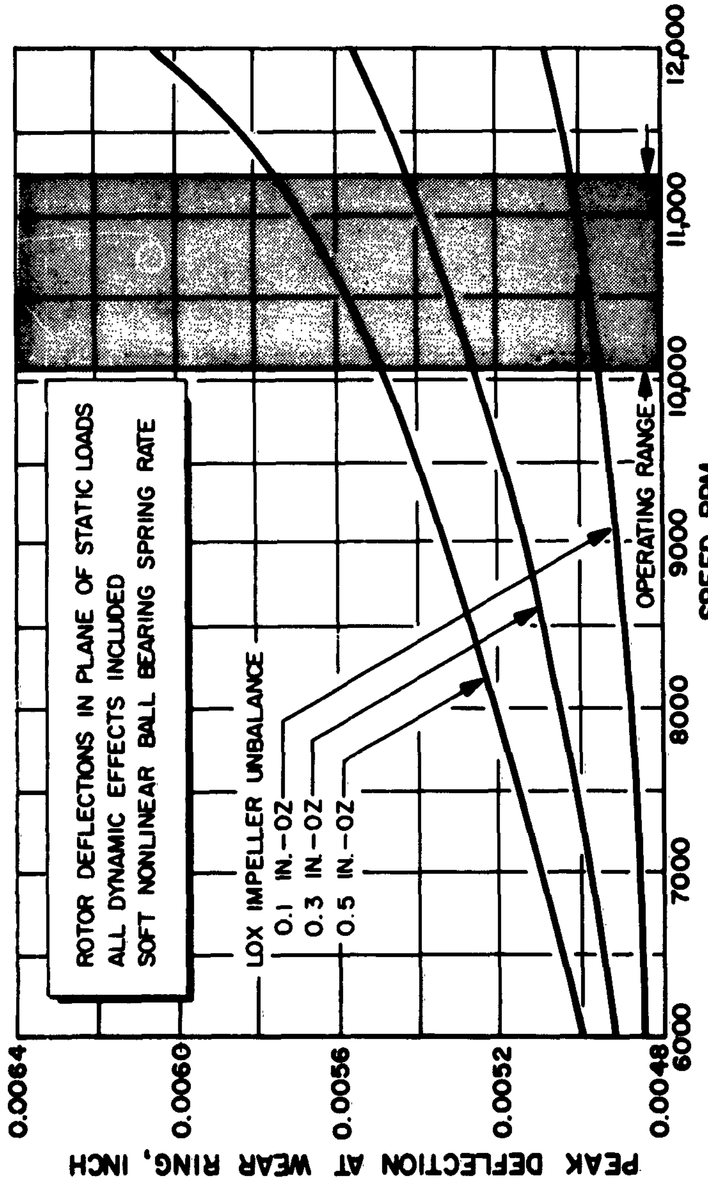

Figure C-9. Rotor Deflections in Plane of Static Loads, Soft Nonlinear Ball Bearing Spring Rate

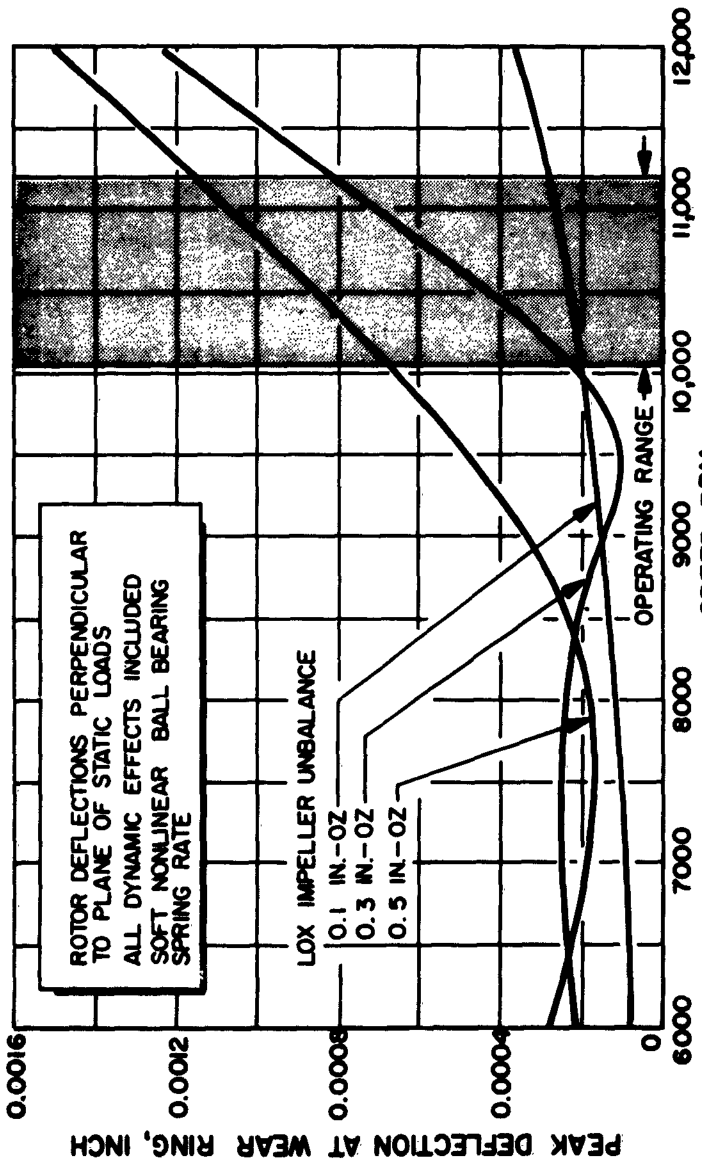

Figure C-10. Rotor Deflections Perpendicular to Plane of Static Loads, Soft Nonlinear Ball Bearing Spring Rate

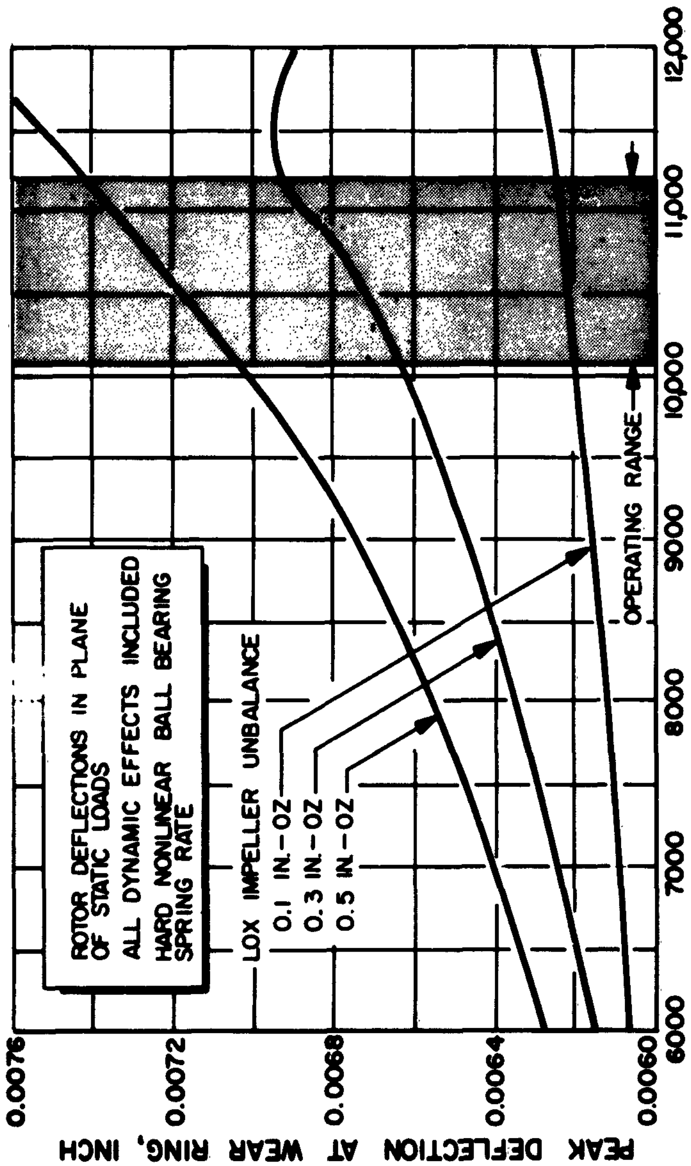

Figure C-11. Rotor Deflections in Plane of Static Loads, Hard Nonlinear Ball Bearing Spring Rate

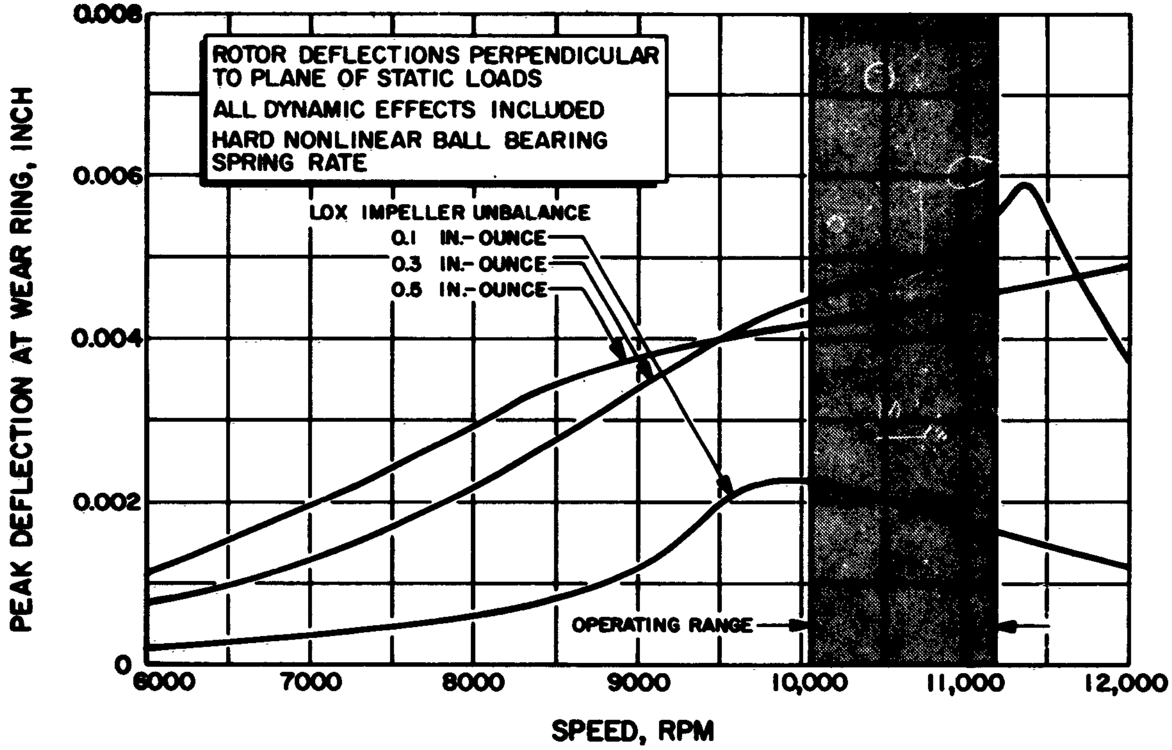

Figure C-12. Rotor Deflections Perpendicular to Plane of Static Loads, Hard Nonlinear Ball Bearing Spring Rate

The inclusion of the inertia of the overhung mass with the equations of motion of the shaft lowered the fundamental critical shaft speed by approximately 8%. Inclusion of the gyroscopic action of the spinning shaft raised the fundamental critical speed by about 4%.

Incorporation of the soft nonlinear ball bearing spring rate did not produce any drastic changes from the linear system response. This was expected since the nonlinearity was rather mild and quite small. With the hard nonlinearity, however, the fundamental frequency of the system could only be excited to any appreciable amplitude as a subharmonic resonance. Thus, the fundamental critical shaft speed was very high for this case.

Several antiresonances and secondary resonances were found for the nonlinear spring rates. Some of these were caused by the presence of the reverse whirling modes of the shaft and others were caused by changes in system parameters caused by increasing deflections. In no case was a secondary resonance sufficiently large to be considered as a critical speed.

The static loading conditions increased the critical speeds of the shaft for the nonlinear systems because of the preset in the springs. These conditions also caused highly elliptical shaft whirling.

The results of the study show conclusively that the critical speed of the Mark 4 shaft is well above the operating speed of the turbopump. Shaft deflections through the turbopump operating speed range were found to be small (less than 0.0064 inch for nominal umbalances).

APPENDIX D

SOLDER CONE TESTS

Analysis of test data following the LOX pump explosion on Atlas missile 1F revealed that the head suppression valve did not open until 0.48 second after sustainer engine start. Subsequent inspection of the turbopump revealed evidence of impeller-to-wear ring rubbing. A 15-test program was initiated and carried out at the Rocketdyne Propulsion Field Laboratory in an attempt to relate these two anomalies. Liquid nitrogen and RP-1 were used as propellants, and MA-3 sustainer engine 2220-1A was used throughout the program. Opening of the head suppression valve was intentionally delayed to simulate the delays experienced with missiles 1F, 2E, and others. The maximum pump shaft deflection was measured during 13 tests; these measurements were made by using solder cones on the wear ring and recording the clearance reduction between the wear ring and impeller.

Special wear rings were prepared with solder cones on the diverter lip and on the wear ring steps. The pump shaft was rotated as the wear ring was assembled, and the solder was worn down to fit the impeller at assembly. The pump was then chilled with liquid nitrogen and the shaft was manually rotated to wear the solder to fit. Measurements were taken before and after childown, giving the dimensional change from this effect. The engine was then started, using the solid-propellant gas generator starter and a controlled engine start sequence. Normal MA-3 start sequence was accomplished during four tests, and actuation of the head suppression valve was purposely delayed during other tests. The head suppression valve was also actuated early to simulate the MA-2/5 start sequence during which the head suppression valve is fully open when pump operation begins. Three tests were conducted with a roughened head suppression valve idler shaft and the bearing cavity intentionally contaminated with water. It was felt that this condition might have existed in the head suppression valve installed on missile 1F. Table D-1 outlines the tests conditions and the test setup. Each test duration was confined to solid-propellant gas generator stage.

The results of this series of tests confirmed that delayed opening of the head suppression valve could have caused the failure of Atlas missile 1F. It was found that delays of the magnitude experienced on missiles 1F and 2F would result in impeller rubbing on the minimum-clearance pumps. Figure D-1 shows that when liquid nitrogen is used as the pump fluid, the delay necessary for rubbing to occur within the minimum clearance pump is approximately 300 milliseconds. The use of LOX as the pump fluid will slightly reduce this time. The tests also revealed that normal MA-3 engine sequencing resulted in a clearance reduction at the wear ring of 0.011 inch at start.

TABLE

SOLDER CONE TEST RESULTS, HEAD SUPPRESSIO

Test No.	Head Suppression Valve Delay from Solid-Propellant Gas Generator Start	Cutoff Type and Time
535-137	45 milliseconds (normal)	Mainstage timer at 3 seconds
-138	479 milliseconds (simulated missile 1F)	Overspeed at 570 milliseconds
-139	500 milliseconds (simulated missile 1F)	Overspeed at 680 milliseconds
-140	939 milliseconds (simulated missile 2E)	Overspeed at 1090 milliseconds
-141	40 milliseconds (normal)	Mainstage timer at 3 seconds
-142	Held closed	Overspeed at 732 milliseconds
-143	930 milliseconds (simulated missile 2E)	Overspeed at 1120 milliseconds
-144	128 milliseconds	Mainstage timer at 3 seconds
-145	40 milliseconds (normal)	Mainstage timer at 3 seconds
-146	126 milliseconds	Mainstage timer at 3 seconds
-147	480 milliseconds (simulated missile 1F)	Overspeed at 670 milliseconds
-148	68 milliseconds (opened valve)	Mainstage timer at 3 seconds
-149*	40 milliseconds (normal)	Mainstage timer at 3 seconds
-150**	85 milliseconds	Mainstage timer at 3 seconds
-151**	130 milliseconds	Mainstage timer at 3 seconds

* The head suppression valve used during this test had (1) water in
(2) a roughened shaft. The purpose was to cause valve hang-up.

**In addition to the conditions of the previous note, these tests w
of 2000 psig compared to 3000 psig used during the previous tests

E D-1

ION VAL

TABLE D-1

T RESULTS, HEAD SUPPRESSION VALVE OPENING INTENTIONALLY DELAYED

Cutoff Type and Time	Maximum Developed Head, Feet of Liquid Nitrogen	Maximum Wear Ring-to-Impe Radial Clearance Reductio	
		Ambient-to-Cold	Cold-t
Mainstage timer at 3 seconds	825	0.008	0.
Overspeed at 570 milliseconds	3125 (occurred as head suppression valve closed)	0.013	0.
Overspeed at 680 milliseconds	2470	0.005	0.
Overspeed at 1090 milliseconds	1890	0.008	0.
Mainstage timer at 3 seconds	717	0.005	0.
Overspeed at 732 milliseconds	2476	0.009	0.
Overspeed at 1120 milliseconds	2400	0.008	0.
Mainstage timer at 3 seconds	1340	0.008	0.
Mainstage timer at 3 seconds	835	Diverter lip remov and clearance incr	
Mainstage timer at 3 seconds	1382	0.006	0.
Overspeed at 670 milliseconds	3731	Diverter lip remov and clearance incr	
Mainstage timer at 3 seconds	440	0.007	0.
Mainstage timer at 3 seconds	...	No measurements mad	
Mainstage timer at 3 seconds	1355	...	
Mainstage timer at 3 seconds	...	No measurements mad	

injected this test had (1) water injected in the idler shaft bearing cavity, and to cause valve hang-up.

were co vious note, these tests were conducted with a hydraulic supply pressure

ts. during the previous tests.

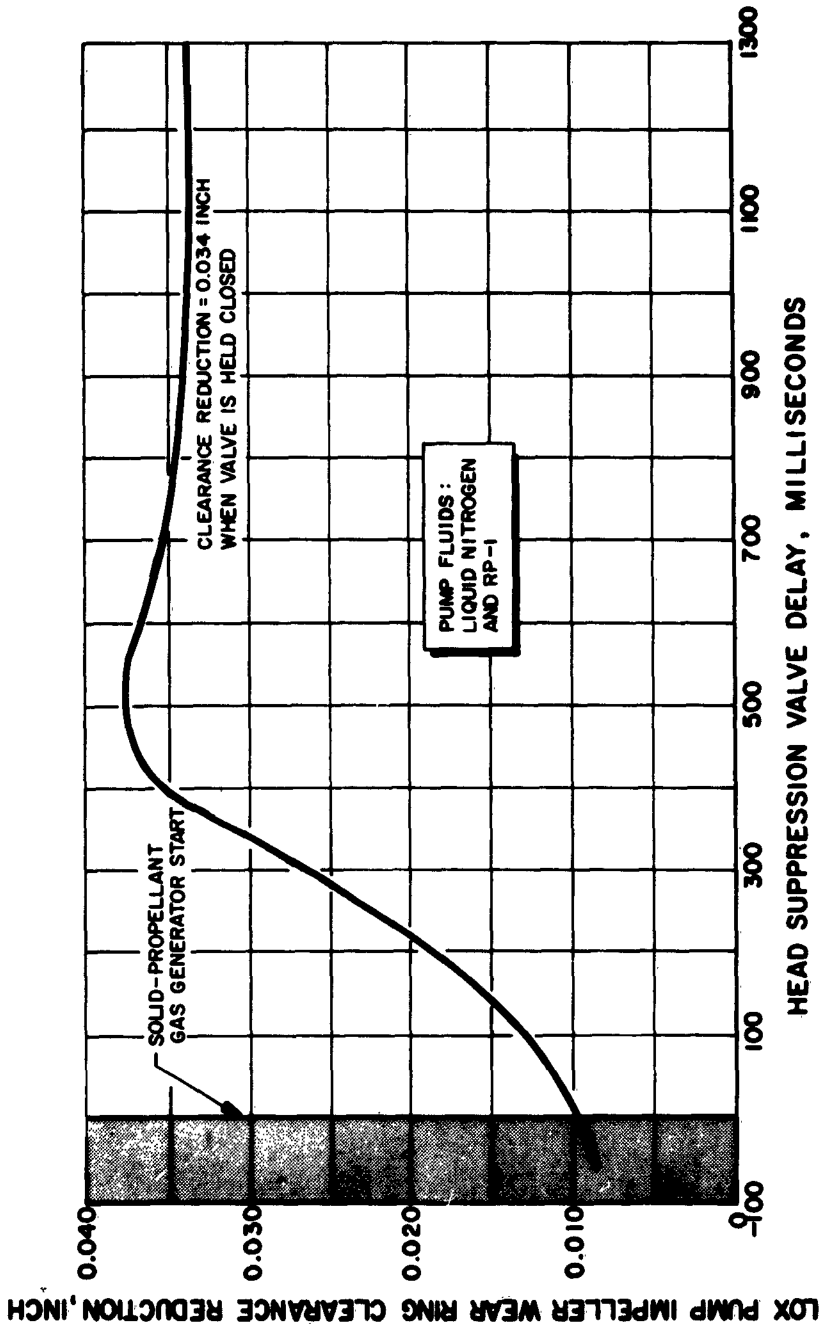

Figure D-1. Impeller Clearance Reduction vs Head Suppression Valve Opening Delay